

Proceedings of:

The Twenty-Ninth Annual Meeting of

The American Society for
Precision Engineering

November 9 to November 14, 2014
The Westin Boston Waterfront
Boston, Massachusetts, USA

The American Society for Precision Engineering (ASPE) is a multidisciplinary professional and technical society concerned with research and development, design, manufacture and measurement of high accuracy components and systems. ASPE activities encompass relevant aspects of mechanical, electronic, optical and production engineering, physics, chemistry, and computer and materials science. Membership is open to anyone interested in any aspect of precision engineering.

Founded in 1986, ASPE provides a focus for a diverse but important community. Other professional organizations have covered aspects of precision engineering, always as a sideline to their principal goals. ASPE is based on the core of generic concepts necessary to achieve precision in any application; independent of discipline, ASPE intends to be the focus for precision technology — and to represent all facets from research to application.

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Preface

This book comprises the proceedings of the 29th ASPE Annual Meeting. The contributions reflect the authors' opinions and are published as presented to ASPE, without change. Their inclusion in this publication does not necessarily constitute endorsement by the ASPE, or its editorial staff.

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Welcome Note

Welcome to the Twenty-Ninth Annual Meeting of The American Society for Precision Engineering. Convening this year in the fabulous city of Boston, Massachusetts, USA, at The Westin Boston Waterfront, the 2014 Annual Meeting will provide a forum for presentation and discussion of the latest technical information and achievements in precision engineering. The meeting will introduce new concepts, processes, equipment, and products while highlighting recent advances in precision measurement, design, control, and fabrication. The program includes a series of oral and poster paper presentations, tutorials on precision engineering topics, exhibits displaying the state-of-the-art in precision engineering products and research, commercial sessions for exhibitor presentations, a student competition, a mentoring session for students and tours of local precision engineering facilities. Many social events including lunches, dinner, breaks and hospitality hours will allow ample time for networking and personal exchanges. If there's time, enjoy the many sites to see in Boston, Cambridge and the New England area.

Precision engineering is important in a wide variety of fields, from manufacturing to microelectronics to basic science. The Society serves as a focus for precision engineers across all of these fields. The Annual Meeting has evolved into an international forum for the exchange of ideas and presentation of research results relating to precision engineering, metrology, controls, and system integration. Precision engineers and scientists from private industry, government laboratories, and universities meet to learn about the latest developments and to exchange ideas about the future directions of these technologies.

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ASPE gratefully acknowledges the time and the effort of the organizing and technical program committee to bring the precision engineering community this program.

29th Annual Meeting

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2014 Scholarship Recipients

2014 Carl Zeiss Student Scholarship



Congratulations to **Herminso Villarraga-Gómez**, University of North Carolina–Charlotte, for being awarded the Carl Zeiss Student Scholarship for 2014. This award includes the ASPE Annual Meeting registration fee and 4 tutorial fees. It also provides for travel and lodging to the ASPE Annual Meeting, as well as to Carl Zeiss Industrial Metrology for a visit to the facility in Maple Grove, Minnesota.

ASPE gratefully acknowledges Carl Zeiss' five year commitment to provide this student scholarship through ASPE. The scholarship is valued at \$5,000 each year.

2014 ASPE Student Scholarships

ASPE is pleased to award 2 student scholarships for the 29th ASPE Annual Meeting. The ASPE Student Scholarships include a waiver of the Annual Meeting registration fee and 4 tutorial fees. It also includes an honorarium to cover travel and lodging expenses to the 29th ASPE Annual Meeting in Boston. This year's scholarship recipients are as follows:

Gregory L. Holst (Craig R. Forest, Advisor), Georgia University of Technology, Georgia, USA
Kazunori Watanabe (Jiwan Yan, Advisor), Keio University, Japan

ASPE is able to award student scholarships because of the generous contributions from organizations, as well as from many ASPE Members and ASPE supporters. ASPE believes the education and support of the next generation of precision engineers is crucial to the advancement of the field.

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Program Schedule

29th ASPE Annual Meeting

Time	Sun., Nov. 9	Mon., Nov. 10	Tues., Nov. 11	Wed., Nov. 12	Thurs., Nov. 13	Fri., Nov. 14
8:00	+ Registration – Harbor Wing		Registration Harbor Wing	Registration – Harbor Wing Student Mentoring Stone Room	Registration Harbor Wing	Technical Tours Meet in Main Lobby of the Westin
9:00	Tutorials* Rooms: Griffin, Lewis, Carlton, Burroughs, Harbor I, Harbor II	Tutorials* Rooms: Griffin, Lewis, Carlton, Burroughs, Harbor I, Harbor II Sightseeing Tour 10:00AM – 4:30PM Meet in Main Lobby of The Westin at 9:45 AM	Technical Session I Harbor Ballroom	Technical Session III Harbor Ballroom	Technical Session VI Harbor Ballroom	
10:00			Break – Galleria	Break – Galleria	Break – Galleria	
11:00			Technical Session II Harbor Ballroom	Technical Session IV Harbor Ballroom	Technical Session VII Harbor Ballroom	
12:00	Lunch On-Your-Own	Lunch On-Your-Own	Lunch and Committee Meetings Marina Ballroom	Awards Lunch and Business Meeting Marina Ballroom		
1:00	Tutorials*	Tutorials*	Commercial Session I Harbor Ballroom	Technical Session V Harbor Ballroom	Technical Session VIII Harbor Ballroom	
2:00			Break – Galleria			
3:00			Commercial Session II Harbor Ballroom	Break – Galleria		
4:00			Poster Session Galleria			
5:00			Poster Session Galleria			
6:00		Welcome Reception Marina Ballroom	Hospitality Hour Galleria	Hospitality Hour Galleria		
7:00					Load Buses to Dinner at the JFK Presidential Library & Museum	
8:00		Keynote Address Harbor Ballroom				
9:00			Meet in Lobby of the Westin			

+ Early Check-In & Registration will be available from 4:00 PM – 7:00 PM on Saturday, November 8 in the Harbor Wing

* Check the hotel event board, the pocket program or the ASPE website for rooms for specific tutorials

 Exhibit Open Hours – Galleria Exhibit Hall

 Student Competition – Stone Room

Keynote Address

Dr. Mark Johnson

U. S. Department of Energy

Monday, November 10, 2014 7:00 p.m.

Session chair: Alexander H. Slocum, Massachusetts Institute of Technology

Innovation in Manufacturing Technologies for Clean Energy



ASPE is pleased to welcome **Dr. Mark Johnson** as this year's keynote speaker. Dr. Johnson currently serves as the Director of the Advanced Manufacturing Office in the U.S. Department of Energy under the Office of Energy Efficiency and Renewable Energy. The Advanced Manufacturing Office (AMO) is focused on creating a fertile innovation environment for advanced manufacturing, enabling vigorous domestic development of new energy-efficient manufacturing processes and materials technologies to reduce the energy intensity and life-cycle energy consumption of manufactured products.

The Keynote Address will take place on **Monday, November 10 at 7:00 PM** at The Westin Boston Waterfront, just after the Welcome Reception.

Technical Exhibitors

ABTech, Inc.	MicroLam
Aerotech, Inc.	Moore Nanotechnology Systems, LLC
Alicona	National Instruments
AMETEK–Precitech, Inc.	New Way Air Bearings, Inc.
attocube Systems AG	nPoint, Inc.
Bal-tec	Optical Perspectives Group LLC
Bilz Vibration Technology	Optikos Corporation
Bruker Nano Surfaces	piezosystem jena
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Chardon Tool	Precision Environments, Inc.
Cranfield Precision, Div. of Fives Landis Ltd.	Precision MicroDynamics, Inc.
Heidenhain Corporation	Professional Instruments Company, Inc.
K&Y Diamond	Riverhawk Company
KAMAN Precision Products	Schneider Optical Machines
KERN Microtechnik GmbH	Starrett Tru-Stone Technologies
Kugler of America, Ltd.	TMC Ametek
Lion Precision	TRIOPTICS USA
Machine Dynamics Research Lab. – Penn State	VDL Enabling Technologies Group USA
Makino	Vermont Photonics Technologies
Market Tech, Inc.	Western Environmental Corporation
	Zygo Corporation

Commercial Session

Tuesday, November 11, 2014

1:30 PM - 2:30 PM

3:00 PM - 4:00 PM

Harbor Ballroom

Session Co-Chairs:

Vivek G. Badami, Zygo Corporation

Seno B. Rekawa, Lawrence Berkeley National Laboratory

Exhibitor representatives are invited to make brief presentations on their organization's products associated with precision engineering. This provides participants with the opportunity to receive timely information on new technologies that have been commercialized into products and services, as well as information on advances that can be expected in the near future.

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2014 ASPE Lifetime Achievement Award



Steven R. Patterson
University of North Carolina–Charlotte

This year's Lifetime Achievement will be awarded to **Dr. Steven R. Patterson**, University of North Carolina–Charlotte. Steve Patterson received his B.S. degree (with honors) in physics from the California Institute of Technology, M.S. and Ph.D. in applied science from the University of California at Davis and M.A. (with honors) in biblical studies from Dallas Theological Seminary. A charter member of ASPE, he has served on its Board of Directors including as president in 2006. He served for more than a decade as an associate editor for Precision Engineering, the journal of the ASPE.

Steve began his involvement with precision engineering in 1970 at the Air Force Weapons Laboratory where he worked with early applications of diamond turning in high-power laser systems, eventually serving there as the Chief of the Optical Physics group. He subsequently joined the Lawrence Livermore National Laboratory as part of the team that designed and built the Large Optics Diamond Turning Machine. At the Lawrence Livermore Laboratory he worked in the areas of fabrication technology and precision interferometry, leading the Materials Fabrication Division there and briefly serving as the initial national program manager for the National Ignition Facility during its conceptual phase. In 1993 he joined the University of North Carolina at Charlotte as the SPX Distinguished Professor of Engineering. For five years beginning in 2003 he returned to the Lawrence Livermore National Laboratory to serve as Associate Director of Engineering before resuming his position at the University.

Steve's technical interests include precision machine design, diamond-turning, machine control and ultra-precision dimensional measurement. He has studied the thermal and long-term temporal stability of a variety of materials used in precision engineering applications. His interests also include engineering education and in particular the training of the next generation of precision engineers.

Steve is a fellow of the SPIE, and has served as a member standards committees for both ASME and IFAC. He is the author or co-author of over fifty papers in the field of optics and precision engineering and a listed inventor on eight patents.

2014 ASPE Distinguished Service Award



Stephen J. Ludwick
Aerotech, Inc.

Dr. Stephen J. Ludwick, Aerotech, Inc., is the winner of this year's Distinguished Service Award. Steve has been a member of ASPE since 1994 and was elected to the ASPE Board of Directors for 2 terms, including as President of the Society. While on the Board as a Director-at-Large, he served as Society Secretary. Steve has served on numerous Annual Meeting and Topical Meeting Committees, has chaired several Topical Meeting Committees, and served as Membership Committee Chair for 4 years. Currently Steve serves as an Associate Editor for the Journal.

Steve is the Director of Mechatronic Research for Aerotech Inc, a manufacturer of precision automation equipment used in advanced manufacturing operations. He joined Aerotech in 1999 and is currently responsible for developing precision motion control systems and feedback control algorithms with an emphasis on the interactions between mechanical, electrical, and algorithmic components of a design. Following a B.S. degree in Mechanical Engineering from Carnegie Mellon University, he studied at the Massachusetts Institute of Technology and received a doctorate in Mechanical Engineering for research into high-dynamic machine tools. Since 2007, he has also taught as an adjunct faculty member at the University of Pittsburgh.

Technical and Poster Sessions

The technical program of the 2014 ASPE Annual Meeting contains over 150 papers on precision engineering advances. Papers that lent themselves to significant verbal interaction, concise but important discoveries, or strongly visual or tactile subjects have been selected for the poster presentation. Authors will be in attendance to discuss their work during both poster sessions on Tuesday, November 11 from 4:00 pm -5:30 pm and Wednesday, November 12 from 3:30 pm -5:00 pm

Session I

Redefining the SI

Tuesday, November 11, 2014, 8:30 AM - 10:00 AM

Session Chair: Byron R. Knapp (Professional Instruments Company) and Christopher J. Evans (University of North Carolina – Charlotte)

- 1. Invited: Contributions from Precision Engineering to the Envisaged New SI**
Bosse, H. (Physikalisch Technische Bundesanstalt) 15
- 2. Invited: A Constant from a Mass, a Mass from a Constant**
Pratt, J. R.; Schlamminger, S.; Chao, L. S. Newell, D. B.; Kubarych, Z.; Abbott, P. (National Institute of Standards & Technology); Cao, A.; Seifert, F.; Haddad, D. (University of Maryland) 20
- 3. The Construction and Characterization of the NIST-4 Permanent Magnet System**
Chao, L. S.; Seifert, F.; Li, S.; Haddad, D.; Schlamminger, S.;
Pratt, J. R.; (National Institute of Standards & Technology) 25
- 4. Fabrication of the Magnetic Circuit for the BIPM Watt Balance**
Ellinghaus, J. M.; Brecher, C.; Wenzel, C. (Fraunhofer Institute for Production Technology)
Arneson, D. A.; Knapp, B. R. (Professional Instruments Company); Kiss, A.; Laverne, T;
Stock, M. (International Bureau of Weights and Measures) 30

Session II

Control of Environment and Vibration

Tuesday, November 11, 2014, 10:30 AM - 12:00 PM

Session Chair: Dannis M. Brouwer (University of Twente) and Dan E. Luttrell (Kriterion, LLC)

- 1. Design and Performance Overview of Advanced LIGO Two-Stage Vibration Isolation and Alignment System**
Matichard, F.; Mason, K. R.; Mittleman, R.; Barnum, S.; Biscans, S.; Foley, S.; Fritschel, P.; Hilliard, M.; Ottaway, D.; Ruet, L.; Sareen, P.; Shoemaker, D.; Stein, A. (Massachusetts Institute of Technology); Abbott, B.; Abbott, S.; Coyne, D.; Macinnis, M. (California Institute of Technology); Allwine, E.; Gray, C.; Grabeel, G.; Kissel, J.; Lhuillier, V.; Paris, H.; Radkins, H.; Robinson, M.; Warner, J. (LIGO Hanford Observatory); Birch, J.; Giaime, J.; Hanson, J.; Le Roux, A.; O'Reilly, B.; Puma, M.; Ramet, C.; Thomas, J.; Vargas, M. (LIGO Livingston Observatory); Lantz, B.; Clark, D.; DeBra, D.; Kucharczyk, C. (Stanford University); DeRosa, R. (Louisiana State University) 35

2. A Multi-DOF Active Vibration Isolation Setup for a Coriolis Mass Flow Rate Meter Staman, K.; van de Ridder, L.; Brouwer, D. M.; van Dijk, J. (University of Twente); Hakvoort, W. B. J. (Demcon)	41
3. Vibration Effects on an Environmentally Tolerant Scanning White Light Interferometer Troutman, J.; Evans, C. J.; Schmitz, T. L. (University of North Carolina at Charlotte)	47
4. Reproducibility and Dynamic Stability of an ABBE-Compliant Linear Encoder-Based Measurement System for Machine Tools Bosmans, N.; Qian, J.; Reynaerts, D. (Katholieke Universiteit Leuven)	51
5. Thermal Error Compensation for Large Heavy Duty Milling-Boring Machines Aguirre, G.; Pérez de Nanclares, A.; Urreta, H. (IK4-IDEKO)	57

Session III

Advances in Precision Metrology

Wednesday, November 12, 2014, 8:30 AM - 10:00 AM

Session Chair: Ralf K. Heilmann (Massachusetts Institute of Technology) and
Jonathan D. Ellis (University of Rochester)

1. Development of a System for 3-D Micro Metrology Using an Ultra-Small Diameter Optical Fiber Probe – Optical Analysis and Evaluative Experiment Using the 0.4 μm Diameter Stylus Shaft. Murakami, H.; Tokuoh, N.; Fukuda, M. (The University of Kitakyushu); Katsuki, A.; Sajima, T. (Kyushu University)	63
2. A Study on Material Influences in Dimensional Computed Tomography Villarraga-Gómez, H.; Morse, E. P.; Hocken, R. J.; Smith, S. T. (University of North Carolina at Charlotte)	67
3. Study on Dynamic Observation of sub-50 nm sized particles in Water Using Evanescent Field with a Compact and Mobile Apparatus Khajornrungruang, P. (Kyushu Institute of Technology); Dean, P. J.; Babu, S. V. (Clarkson University)	73
4. Installation of a Vibrating Contacting Probe onto an Ultra Precision CMM Claverley, J. D.; Leach, R. K. (National Physical Laboratory); Baas, M.; Widdershoven, I.; Spaan, H. A. M. (IBS Precision Engineering)	78
5. Machined Workpiece Error Prediction Callaghan, R. P. (Independent Quality Labs, Inc.); Charlton Jr., T. (Charlton Associates, LLC)	84

Session IV

Precision Motion Control

Wednesday, November 12, 2014, 10:30 AM - 12:00 PM

Session Chair: Stuart T. Smith (University of North Carolina at Charlotte) and
Shih-Chi Chen (The Chinese University of Hong Kong)

1. Friction Parameter Identification and Compensation Using the Elastoplastic Friction Model Keck, A.; Zimmermann, J.; Sawodny, O. (University of Stuttgart)	89
2. Data-Driven Control Strategy for a Reticle Stage in a Lithographic Tool Jiang, Y.; Yang, K.; Zhu, Y.; Hu, C. X. (Tsinghua University)	95

3. Hysteresis Motor Driven One Axis Magnetically Suspended Reaction Sphere Zhou, L.; Imani Nejad, M.; Trumper, D. L.; (Massachusetts Institute of Technology)	101
4. Large-Stroke Piezo-Actuated Planar Motor For Nanopositioning Applications Zhang, J.; Zou, H.; Xiong, Z. (University of Michigan-Shanghai Jiao Tong University Joint Institute); Huang, P. (Shanghai Jiao Tong University)	105
5. Tracking Control Performance of Ultrahigh Acceleration and High Velocity Linear Drive Mechanism Using Learning Control Method Sato, K.; Hama, T. (Tokyo Institute of Technology)	111

Session V

Ultra-Precision Machining and Forming

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Session Chair: Alexander H. Slocum (Massachusetts Institute of Technology) and
Craig R. Forest (Georgia Institute of Technology)

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2. Soft Mold-Based Hot Embossing for High-Resolution Patterning on Nonplanar Surfaces Chen, J.; Chen, S.-C (The Chinese University of Hong Kong)	121
3. Design and Development of High-Frequency Actuator for Nanocoining Nowak, W. J.; Garrard, K. P.; Dow, T. A. (North Carolina State University).	126
4. Synchronization System of Galvanometer Scanner and 2-axis Linear Stage for Laser Routing in Wide Area FPCB coverlay Kim, K.; Lee, J.; Yoon, K.; Park, J. (Korea Institute of Machinery & Materials (KIMM))	132
5. Fabrication of a Flexure-based Linear Translation Stage with Integrated Optical Position System Bado, P.; Said, A. A. ; Dugan, M. A. (Translume); Bellouard, Y. (Eindhoven University of Technology)	134

Session VI

Design of Precision Machines

Thursday, November 13, 2014, 8:30 AM - 10:00 AM

Session Chair: Mark T. Kosmowski (Electro Scientific Industries) and
Bradley H. Jared (Sandia National Laboratories)

1. Mechanical Design of Multiple Fresnel Zone Plates Precision Alignment Apparatus for Hard XRay Focusing in Twenty-Nanometer Scale Shu, D.; Liu, J.; Gleber, S.-C.; Vila-Comamala, J.; Lai, B.; Maser, J.; Roehrig, C.; Wojcik, M. J.; Vogt, S. (Argonne National Laboratory)	138
2. Actuator Design for a High-Speed Large-Range Tip-Tilt-Piston Micro-Mirror Array Panas, R. M. (Lawrence Livermore National Laboratory); Hopkins, J. B. (University of California - Los Angeles).	142
3. Flexure Design for a High-Speed Large-Range Tip-Tilt-Piston Micro-Mirror Array Hopkins, J. B. (University of California - Los Angeles); Panas, R. M. (Lawrence Livermore National Laboratory)	147

4. **Design of a Flexure-Based Dynamic-Tunable Five-Axis Nanopositioner**
Li, C.; Wang, J.; Chen, S.-C. (The Chinese University of Hong Kong) 153
5. **Design of a Microfluidic Device to Extract Dyes from Fibers for Forensic Analysis**
Gunning, S. P.; Garrard, K. P.; Furst, S. J.; Dow, T. A. (North Carolina State University). 158

Session VII

Interferometry and Optics

Thursday, November 13, 2014, 10:30 AM - 12:20 PM

Session Chair: Marcin B. Bauza (Carl Zeiss Industrial Metrology, LLC) and
Deming Shu (Argonne National Laboratory)

1. **Invited: Interferometry and Optics as Employed in Advanced LIGO**
Shoemaker, D. H. (Massachusetts Institute of Technology). 162
2. **Wide-Field Interference Microscopy for Areal Topography of Precision Engineered Surface**
Biegen, J. F.; Colonna de Lega, X.; de Groot, P. J. (Zygo Corporation) 163
3. **Development of a Miniature, Multichannel, Extended Range Fabry-Perot Fiber-Optic Laser Interferometer System for Low Frequency Si-Traceable Displacement Measurement**
Nowakowski, B. K.; Smith, D. T. (National Institute of Standards & Technology);
Smith, S. T. (University of North Carolina at Charlotte) 167
4. **A Novel Heterodyne Grating Interferometer System for In-Plane and Out-of-Plane Displacement Measurement with Nanometer Resolution**
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5. **Phase Compensation for Dynamic Doppler Frequency Shifts**
Wang, C.; Ellis, J. D. (University of Rochester). 178

Session VIII

Precision Manufacturing and Assembly

Thursday, November 13, 2014, 1:30 PM - 3:00 PM

Session Chair: Jonathan B. Hopkins (University of California - Los Angeles) and
Mark A. Stocker (Cranfield Precision, Division of Cinetic Landis Ltd.)

1. **A Comparative Performance Analysis of FDM Machines Based on a Calibration Artefact**
D'Angelo, G.; Hansen, H. N.; Nielsen, J. S.; Rasmussen, J.;
Pedersen, D. B. (Technical University of Denmark) 183
2. **Micro-Laser Assisted Single Point Diamond Turning on Unpolished Single Crystal Silicon**
Mohammadi, H.; Poyraz, H. B.; Patten, J. A.; (Western Michigan University);
Ravindra, D. R. (Micro-Laser Assisted Machining Technologies, LLC) 188
3. **Efficient Micro-Concentrator for Microsystems-Enabled Photovoltaics**
Jared, B. H.; Saavedra, M. P.; Anderson, B.; Miller, B.; Zamora, D. L.; Sweatt, W. C.;
Nielson, G.; Okandan, M.; Elisberg, B. (Sandia National Laboratories) 193
4. **Design and Fabrication of an Optical System for a Balloon-Borne Space Telescope**
Furst, S. J.; Dow, T. A.; Garrard, K. P. (North Carolina State University). 196

5 Analysis of Nozzle Design Used for the Creation of Advanced Energy Beam	
Yu, N.; Jourdain, R. P.; Gourma, M. (Cranfield University); Shore, P. (Loxham Precision Limited).	200

Poster Sessions

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Technical Papers

CONTRIBUTIONS FROM PRECISION ENGINEERING TO THE ENVISAGED NEW SI

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INTERNATIONAL SYSTEM OF UNITS

The system of Weights and Measures was recognized as a necessary basis for science, agriculture, construction and trade from the very beginning of ancient civilization. The importance of a reliable system of Weights and Measures or generally speaking of a reliable System of Units is documented by the fact that it has always been regarded as a governmental task of high priority until today.

In 1960 the 11th General Conference on Weights and Measures (CGPM) adopted the name *Système International d'Unités* (International System of Units, international abbreviation SI), for the recommended practical system of units of measurement. The SI base units are a choice of seven well-defined units which by convention are regarded as dimensionally independent: the metre, the kilogram, the second, the ampere, the kelvin, the mole, and the candela. Derived units are those formed by combining base units according to the algebraic relations linking the corresponding quantities [1]. Taking into account the historical development of the system of units, also the SI was thought not to be static but to evolve to match the world's increasingly demanding requirements for precise measurements.

One important change to the SI was made in 1983 when the unit of length, the metre was defined by fixing the - at this time - best known numerical value of a natural constant, the speed of light in vacuum, namely, 299 792 458 metre per second.

ENVISAGED "NEW SI"

This concept of defining an SI base unit by fixing the numerical value of a properly chosen natural constant or at least a so-called defining constant also inspired the envisaged "New SI" or the possible future revision of the SI.

In the "New SI" four of the SI base units, namely the kilogram, the ampere, the kelvin and the mole, will be redefined in terms of invariants of

nature; the new definitions are proposed to be based on fixed numerical values of the Planck constant (h), the elementary charge (e), the Boltzmann constant (k), and the Avogadro constant (N_A), respectively, see figure 1.



FIGURE 1. Scheme of the base units and the defining constants in the New SI.

Different highly sophisticated experiments have been performed to realize precise measurements of the above mentioned natural constants. However, "while remarkable progress has been made over the last few years, the conditions for adopting the redefinitions, as set by the CGPM at its 23rd meeting (2007), have not yet been fully met" [2].

It is important to note, that *progress in precision manufacturing and precision engineering* is necessary to enable the envisaged New SI. In this contribution some examples of precision machining tasks and achievements in precision dimensional metrology will be discussed.

The most demanding conditions are set for the determination of the Planck constant h , which in the New SI will be used to define the unit of mass, the kg. The Consultative Committee for Mass (CCM) has formulated different prerequisites [3] before the new definition of the unit of mass could be put into place as well as a roadmap of different necessary steps to be achieved to reach the target until 2018. The CCM roadmap activity has been endorsed by the CIPM at its 102nd meeting in 2013 [4].

One of the conditions defined by the CCM is that at least three independent experiments, including work from watt balance and XRCD experiments, yield consistent values of the Planck constant with relative standard uncertainties not larger than $5 \cdot 10^{-8}$ and that at least one of these results should have a relative standard uncertainty not larger than $2 \cdot 10^{-8}$.

PRECISION EXPERIMENTS FOR h

To determine the value of the Planck constant with the above mentioned uncertainties two different approaches are followed in metrology institutes worldwide, namely the watt balance experiments and the x-ray crystal density (XRCD) experiments [5].

In the watt balance experiments the gravitational force of a mass standard is compared with compensating electromagnetic forces generated by a coil in a static homogenous magnetic field in two different configurations, namely with a stationary coil carrying a defined current (weighing phase) and with the same coil moving at constant speed, thus inducing a constant voltage over the coil (moving phase).

The precise alignment of the components of the watt balance as well as the control and interferometric measurement of the linear movement of the coil in the dynamic phase of the experiment are very demanding precision engineering challenges related to the watt balance experiments. In [6] an analysis on the alignment requirements was given for the watt balance operated at NRC. Different influences were analyzed, such as alignment of the laser beam, residual torques about horizontal axes and horizontal forces, alignment of the mass pan of the balance, determination of the Abbe offset, measurement and adjustment of the horizontal velocities, measurement of the angular velocities and changes in the coil position between phases. The resulting combined relative alignment uncertainty was estimated to be $5.5 \cdot 10^{-9}$. Recently, a new measurement value for the Planck constant h using the NRC watt balance has been reported with a relative uncertainty of $1.8 \cdot 10^{-8}$ [7], which would satisfy the condition set by CCM with respect to the smallest uncertainty of one from at least three independent experiments to be not larger than $2 \cdot 10^{-8}$.

In the XRCD method one uses a high purity, highly enriched ^{28}Si sphere (diameter of 93 mm

and mass of about 1 kg) to precisely determine the Avogadro constant (N_A). The Avogadro constant links the atomic to the macroscopic world and is determined by the ratio of the density of an atomic unit cell of the ^{28}Si sphere and the density of the macroscopic sphere.

The measurement of the Avogadro constant is based on $N_A = n \cdot M / (\rho \cdot a^3)$, where $n = 8$ is the number of atoms per unit cell of a silicon crystal and ρ , M and a are the density, molar mass and lattice parameter, respectively. Precise measurements of the molar mass, the Si lattice parameter as well as the mass and the volume of the Si sphere are thus required to determine N_A . Because the molar Planck constant $h \cdot N_A$ is known from spectroscopy experiments with an uncertainty of $7 \cdot 10^{-10}$ [8], a precise measurement of N_A also provides a precise value of h . The most precise value for N_A using the XRCD method with a standard uncertainty of $3 \cdot 10^{-8}$ was published by the international Avogadro consortium in 2011 [9].

In addition to the production requirements for the pure isotope enriched silicon single crystal material, also the manufacturing specifications for surface quality (average roughness values below 0.3 nm) and spherical form (roundness deviation below 30 nm in amplitude) of the macroscopic silicon spheres are very demanding [10].

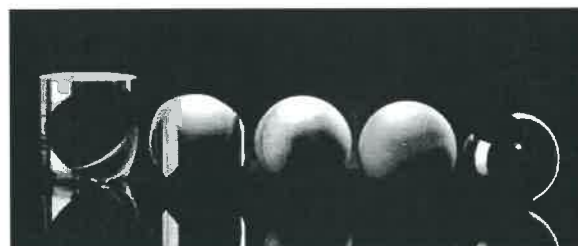


FIGURE 2. Overview of different sphere manufacturing steps (hollowed ingot, cut form, turned, coarsely lapped and polished sphere).

The biggest uncertainty contribution - about 2/3 of the total budget - so far is due to the volume determination of the silicon spheres. A special spherical Fizeau interferometer has been developed for this purpose, see figure 3 for a schematic drawing [11] as well as techniques to characterize the influence of different surface layers on the determination of N_A [12].

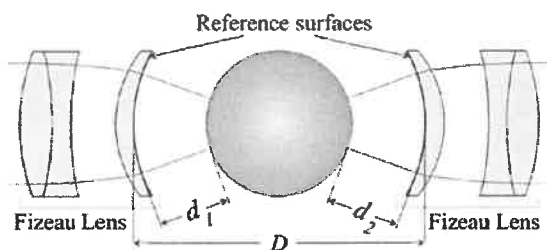


FIGURE 3. Schematic drawing of the central part of the interferometer. The measured quantities are the distances between the sphere surface and the reference surfaces, d_1 and d_2 , and the length of the empty etalon D .

Figure 4 shows a graphical representation of the resulting radius topography from all measurements performed on one of the silicon spheres manufactured along the process chain referred to in figure 2. One can clearly recognize the high symmetry with the characteristic of a rhombic dodecahedron – in the magnitude of several nanometers. This typical appearance of a cubic crystal is worked out in a unique and clear form (see figure 5) and could also be repeated on other spheres.

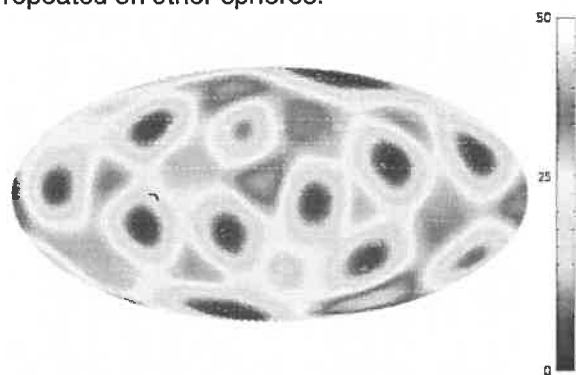


FIGURE 4. Radius topography of silicon sphere PTB_11-01.

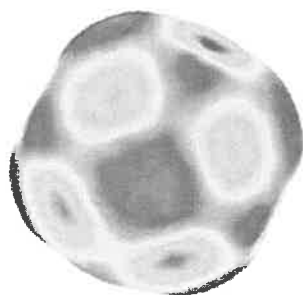


FIGURE 5. Deviations from roundness of sphere PTB_11-01, 10^6 -fold enhanced.

Although today three independent experiments for h have been published with uncertainties below $5 \cdot 10^{-8}$ and one experiment with an uncertainty below $2 \cdot 10^{-8}$ which also seem to be fairly consistent, one should not forget, that the CCM also requires procedures for the future realization and dissemination of the kilogram, as described in the *mise en pratique*, have been validated in accordance with the principles of the CIPM – MRA. This validation process of the procedures will need additional time as foreseen in the CCM roadmap.

Figure 6 shows up-to-date results and standard uncertainties of different experiments (watt balances and Avogadro) for determination of h , as published in [13].

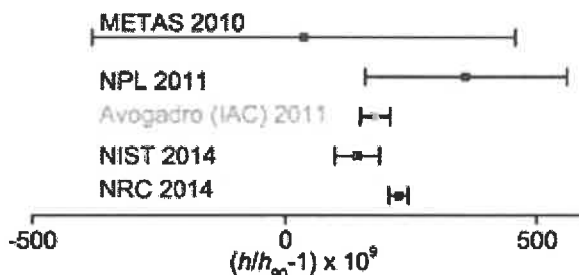


FIGURE 6. Recent measurement results for Planck's constant h [13].

PRECISION EXPERIMENTS FOR k

For precise determination of the Boltzmann constant k , different approaches are followed at NMI's worldwide. The most promising method with smallest achievable relative standard uncertainties below 10^{-6} is called acoustic gas thermometry (AGT). One experiment uses acoustic waves in a noble gas inside a precisely manufactured slightly ellipsoidal resonator to determine k [14], see figure 7.

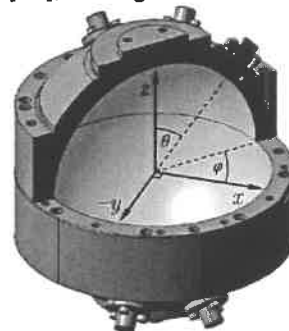


FIGURE 7. Manufactured resonator for the NPL Boltzmann experiment [14].

The small uncertainty obtained with this experiment was reported to be $u_R = 0.71 \cdot 10^{-6}$. In addition to other AGT experiments using spherical or slightly ellipsoidal resonators, there is another experiment which uses precisely manufactured cylindrical resonators for the AGT experiment [15].

One of the other independently developed methods for determination of k is based on dielectric-constant gas thermometry DCGT [16]. Here, the dielectric constant of a noble gas is measured under different pressures using a precise pressure balance. The calibration of the pressure balance is traceable to dimensional characterizations (diameter and form) of the effective area of cylindrical piston-cylinder pressure gauges, as shown in figure 8. Standard uncertainties of three-dimensional data of 8 nm for pistons and 16 nm for cylinders were obtained by a combination of high precision form and diameter measurements [17].



FIGURE 8. Piston cylinder pressure gauges used for the determination of the Boltzmann constant k by means of the DCGT method.

SUMMARY AND OUTLOOK

Some of the contributions from precision manufacturing and precision engineering for the progress of experiments in fundamental metrology aiming at the envisaged New SI were addressed. It has been shown how progress in manufacturing chains as well as in alignment and characterization of critical components of experiments contribute to the required further reduction of uncertainty of different experiments for determination of more precise numerical values of natural constants needed for the proposed definitions of the New SI. Although today the formal requirements of the CCM with respect to the uncertainty and consistency of at least three independent experiments for determination of the Planck's

constant h seem to be fulfilled, it will be of importance to follow the results of other watt balance experiments as well as repeated Avogadro experiments with a second lot of ^{28}Si material, which are under preparation.

It is also of importance to maintain the established manufacturing chains and the described precision experiments for future use.

ACKNOWLEDGEMENTS

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A constant from a mass, a mass from a constant

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Introduction

The International System of Units (SI) will soon be based on a set of fundamental constants with fixed, exact values known as defining constants [1,2]. For example, the unit of length was once derived from the space between two scratches on a metal bar, but now it can be derived anywhere at any scale using a defining constant, the speed of light. Similarly, it is planned that the unit of mass will cease to be derived from a single cylinder of platinum iridium (Pt-Ir) in a vault in Sevres, France near Paris, and instead it may be derived anywhere at any scale using the defining constants, most notably a fixed value of the Planck constant h .

The first step, of course, is fixing a value of h , which is now defined inexactly in terms of the existing standard of mass, the International Prototype of the Kilogram (IPK). This must be done based on experiments, with an uncertainty of less than 5 parts in 10^8 if the precision of mass is to be preserved after revising the units of measure. Multiple groups have recently crossed this threshold in precision using watt balance [3,4] and x-ray crystal density [5] approaches. At last, the long discussed decision whether to continue deriving h from the International Prototype of the Kilogram, or to instead derive the unit of mass from a fixed value of h [6], appears to be drawing to a close. A timeline has emerged: a pilot study on the realization of the kilogram from h is proposed for 2015, with revision of the SI slated for 2018.

The National Institute of Standards and Technology (NIST) recently used a watt balance known as NIST-3 to measure the Planck constant in terms of IPK with a relative uncertainty of approximately 4.5 parts in 10^8 [3]. Along the way to this new NIST value of h , the instrument was also employed to perform the

reciprocal experiment: h was “fixed” and the unknown mass of a stainless steel mass standard was measured with reference only to standards of length, time (frequency), and electrical quantities, all derivable from fixed fundamental constants. This paper reviews the basic principles of a watt balance experiment and shares the results of this trial dissemination of mass directly from a set of defining constants, rather than from an artifact.

Watt Balance Principles

As illustrated in Figure 1, a watt balance has many elements in common with an ordinary compensation balance. There is a balance mechanism (a wheel pivoting on a knife edge in this case), a weighing pan suspended from the balance, and a moving coil actuator to apply compensation forces to hold a null position.

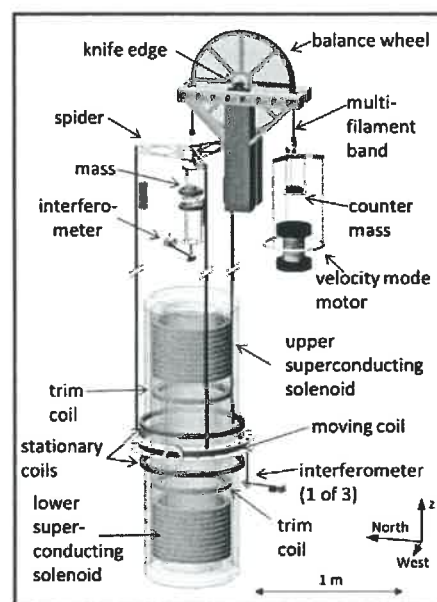


Figure 1 Schematic drawing of the NIST-3 watt balance.

In operation, such balances employ the substitution principle. The force on the mass pan is either generated by the gravitational force from the mass set on top of the mass pan or by an electromagnetic force created by the actuator located below the mass pan. However, what sets a watt balance apart from its compensation balance counterpart is its ability to operate alternately in two separated modes.

In the weighing mode, dc current I is fed back through a coil in a magnetic field to produce a force that nulls the position of the balance mechanics and exactly balances the gravitational force acting on the test mass m , so that $BLI=mg$, where B is the magnetic flux density, L is the wire length in the coil, and g is the local gravitational acceleration.

In the velocity mode, the geometric factor BL is precisely calibrated in terms of length, time, and electrical standards. The calibration is achieved by moving the coil through the magnetic field while measuring its velocity v and the voltage U_v induced across the coil, i.e., $U_v=BLv$.

Measurements from the two operational modes allow one to compute a virtual comparison of electrical power to mechanical power, $U_v I = mgv$, both measured in watts, hence the name watt balance. A small number of such balances have been constructed over the years, and more are in the planning stages, as evident in the recent special issue of the journal *Metrologia* [7]. In the next section, we will focus on the interesting connection the experiment makes between classical and quantum physics.

A constant from a mass

The virtual comparison of electrical power to mechanical power embodied in the watt balance relation $U_v I = mgv$ creates an opportunity to evaluate a fundamental constant of quantum physics in terms of a Newtonian quantity, gravitational mass. This opportunity arises because the references used to measure the induced voltage U_v and the applied current I are very precisely known in terms of the Planck constant.

The basis of modern voltage standards is the Josephson effect, named after Brian Josephson, who was awarded the 1973 Nobel prize in physics for its discovery in 1962 [8]. The Josephson effect describes the quantum

mechanical behavior of electrons in a pair of superconductors separated by a thin, non-superconducting barrier. Taken as a whole, this superconductor sandwich is called a Josephson junction. The electrons in the superconducting elements of these junctions are characterized by quantum mechanical wavefunctions, the relative phases of which can be locked through interaction with electromagnetic waves (i.e., by exposing the junction to microwave radiation). Because of this phase locking of the wavefunctions, a fixed voltage $V = fnh/2e$ develops across an array of such junctions, where f is the frequency of the electromagnetic excitation, n is the number of Josephson junctions in the array, e the elementary charge, and h the Planck constant. The summed junction voltage is a quantum reference that is stable and repeatable to a few parts in 10^{10} and depends only on a frequency, everything else being a constant.

During the velocity mode of watt balance experiments performed at NIST, the voltage induced across the coil U_v is compared to a fixed reference voltage produced by a programmable array of such Josephson junctions that was developed by another group of researchers at NIST [9]. The NIST Programmable Josephson Voltage System works at a selectable microwave frequency and features software controls that allow the user to program the array of junctions to produce a variety of reference voltages, including limited waveform capabilities. In practice on the NIST watt balance, the system is used to generate a nominal 1 V reference. The velocity of the balance is then controlled to minimize the difference (error signal) between U_v and this reference as measured using a bridge type circuit arrangement. This establishes U_v in terms of f , n , and the ratio $1/K_J$ where $K_J = 2e/h$ is referred to as the Josephson constant.

A second quantum mechanical phenomenon is necessary to completely link a watt balance to the Planck constant. This phenomenon is a special variant of the Hall effect known as the *quantum* Hall effect. Recall that the Hall effect describes the migration of charge carriers in a conductor to one side of the conductor in the presence of a perpendicular magnetic field. As a result, a voltage appears across the conductor in a direction perpendicular to both the current flow and magnetic field direction. In 1985, Klaus von Klitzing was awarded the Nobel Prize in physics

for his experimental work showing that this Hall voltage becomes exactly quantized if the conductor is a two dimensional electron gas in the presence of a strong magnetic field [10]. The energy level of the electrons and hence the Hall resistance is quantized in this case at integer fractions of $R_K = h/e^2$, where R_K is known as the von Klitzing constant. At present, NIST uses a gallium arsenide heterostructure cooled to 0.3 K in order to create a two dimensional electron gas suitable as a quantum Hall resistance standard and to serve as the US representation of the ohm [11]. NIST has also made great progress recently in the development of graphene based quantum Hall devices [12] that show potential of working at higher temperatures and lower magnetic fields than available with present devices.

During the weighing mode of a watt balance experiment, the electric current to null the balance position as the test mass is placed on and off the balance is measured using Ohm's law by passing the current through a resistor and measuring the corresponding voltage drop, U_w . The value of U_w is once again measured via comparison to a Josephson array voltage standard, while the value of the resistor, R , is known, having been determined via comparison to a quantum Hall standard, $R = \alpha R_K$, where α is a proportionality constant measured using state-of-the-art bridge techniques [11].

With data available from both modes of the experiment, the electrical power is written as $P = UI = U_v U_w / R$. Because both of the voltages are measured via the Josephson effect, and the resistance is measured via the quantum Hall effect, one obtains for the electrical power

$$P = n_w n_v f_w f_v h / 4\alpha$$

In order to make a measurement of h , the mass m of a stable, Pt-Ir mass standard is determined by comparison to the US National prototype kilogram, then the local gravitational acceleration g is measured with an absolute gravimeter [13] and the electrical and mechanical powers are equated and solved for h

$$P = mgv \text{ or } h = 4\alpha mgv / (n_w n_v f_w f_v).$$

A value of $h = 6.626\,069\,79 \times 10^{-34} \text{ J s}$ was determined over the course of six measurement campaigns in 2013 using a platinum iridium

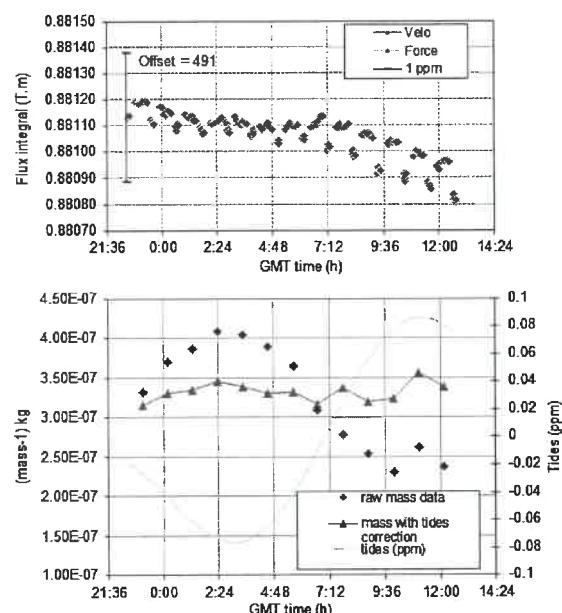


Figure 2. Upper plot is typical raw velocity mode and weighing mode data over a twelve hour period. Lower plot shows the magnitude of the relative variation of gravity due to tidal effects and the measured deviation of the mass before and after correction for this variation.

prototype. Typical data used for this determination are shown in Figure 2.

A mass from a constant

The equation for h in terms of mass can be rearranged to solve for mass in terms of h , or

$$m = n_w n_v f_w f_v h / (4\alpha g v),$$

where m , in this case, refers to the mass of an unknown test artifact to be determined. It is interesting to note that in this scheme, the unit of mass is no longer tied to a single cardinal value. The impact on scaling the unit of mass is beyond our scope here, but as observed in a recent essay by the lead author [14] it opens up the possibility of accurately weighing everything from "atoms to apples" in terms of the defining constants.

In June 2013, a stainless steel mass with a value unknown to the researchers at the watt balance was used on the NIST-3 watt balance. After 25 days of measurement on the watt balance, the mass of the artifact was determined using a previously recorded value $h = 6.626\,069\,79 \times 10^{-34} \text{ J s}$ to be

$$m_{WBV} = (1 \text{ kg} + 338 \mu\text{g}) \pm 60 \mu\text{g}$$

The above estimate of the object's mass was determined in vacuum (0.5 Pa), in the absence of any significant buoyant forces. The value includes corrections for a variety of factors, including tidal corrections to gravity (e.g., see [3]). An important correction new to this work was the effect of magnetic forces arising from the weak magnetic properties (i.e., susceptibility and magnetization) of the stainless steel artifact.

Briefly, the magnetic properties of this artifact were determined using the balance apparatus. Changes in force on the balance were recorded as the magnetic flux densities around the balance pan were varied in a systematic fashion. Prescribed fields were applied using auxiliary coils positioned around the mass pan. From these force measurements, the volume magnetic susceptibility and the polarization of the artifact were deduced. The magnetic flux density at the weighing position due to the superconducting magnet and the vertical gradient of its field at this position were also determined. Combining these observations with an estimate of the artifact volume, the magnetic field was found to contribute an additional apparent mass of $11.6 \mu\text{g} \pm 6 \mu\text{g}$. This correction was already applied to the vacuum mass value given above.

Finally, the true mass of the artifact in air is slightly heavier than the vacuum mass, since an adsorbed layer of contaminants (mostly water) clings to mass artifacts when they are used in air. Because of this, we must explicitly estimate this sorption correction if we wish to compare the watt balance value to the true mass of the artifact that will be measured using conventional practices in air.

Based on data available in reference [15], which examined sorption factors for Pt-Ir and stainless steel kilogram mass artifacts using a vacuum mass comparator, we estimate the amount of material adsorbed on the surface to be $\approx 14 \mu\text{g} \pm 5 \mu\text{g}$, so that the true mass, as determined by the watt balance approach, is

$$m_{WB} = (1 \text{ kg} + 352 \mu\text{g}) \pm 60 \mu\text{g}$$

In contrast, before and after the watt balance experiment, the NIST Mass and Force Group measured this stainless steel artifact on a mass

comparator at ambient air conditions using a weighing design that included the Pt-Ir US National standard, for a determination of the true mass in terms of the SI kg. This value, m_{MG} , includes a correction for the buoyant force of air acting on the artifact. In the most basic weighing comparison [16], this correction appears as follows

$$m_{MG} = m_{PtIr} - \rho_a(V_{PtIr} - V_{SS}) - C$$

where m_{PtIr} is the mass of the US National standard, ρ_a is the density of air, V_{SS} and V_{PtIr} are the volume of the unknown stainless steel and the platinum iridium US National standard mass artifacts, respectively, and C is the balance reading. Observe that this determination of mass includes the mass of adsorbed contaminants implicitly, since the entire determination was performed in ambient air. The magnitude of such contamination (in terms of mass) can vary over time, depending on a variety of features of the metal surface and its surrounding ambient environment, including obvious factors such as the surface roughness of the artifacts and the chemical composition of the metal surfaces and the air.

The NIST Mass and Force Group obtained the following value of the true mass for the stainless steel artifact

$$m_{MG} = (1 \text{ kg} + 324 \mu\text{g}) \pm 14 \mu\text{g}.$$

The difference between this value, and that determined on the watt balance is

$$m_{WB} - m_{MG} = 28 \mu\text{g} \pm 62 \mu\text{g}.$$

No significant difference between the two measurement methods is observed. This demonstrates the utility of a watt balance for making direct calibration of the true mass of stainless steel mass artifacts, which are the most common type of standards employed for mass metrology.

Conclusion

The SI is poised for a major revision where the unit of mass will no longer be based on an artifact definition. The experiment described here illustrates that, as expected, a watt balance capable of measuring the Planck constant can be used effectively in a reciprocal mode, so that the realization of mass at the kilogram level is

possible. In this regard, the experiment provides a useful preview of the methods and issues that surround the planned revision of the definition of the SI unit of mass. The experiment highlights the close coordination that is required between artifact mass metrology and the more abstract realization of the unit of mass engendered by the watt balance experiment, particularly during this transition from one unit definition to the other. That both methodologies yielded here the same outcome should lend confidence to the notion that a new definition of mass based on defining constants can succeed. Moving forward, it is apparent that the coordination between artifact metrology and watt balance experimentation must persist. After all, artifacts calibrated through reference to defining constants using a watt balance will be the means of disseminating the unit beyond NIST. Such artifacts are fully compatible with the existing mass infrastructure, and, until such time as watt balance instruments proliferate and become commonplace, traceability to the handful of such realizations will be achieved as it has always been, through exchanges of artifact mass standards.

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THE CONSTRUCTION AND CHARACTERIZATION OF THE NIST-4 PERMANENT MAGNET SYSTEM

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ABSTRACT

A watt balance is an electromagnetic force balancing instrument to realize the unit of mass at the kilogram level. The magnet system is one of the key components. Our group at the National Institute of Standards and Technology is currently building a next-generation, permanent-magnet driven watt balance, NIST-4. We describe the construction of the magnet system, characterization of the field profile, and adjustment methods used to achieve an ideal magnetic field profile. Although the absolute field strength is not critical, the uniformity of the magnetic flux profile through the air gap of the magnet is essential for a watt balance experiment.

INTRODUCTION

A redefinition of the International System of Units, the SI, is impending and may occur as early as 2018. Specifically, in the context of mass metrology, the base unit kilogram will be redefined in terms of Planck's constant, severing its current ties to the International Prototype Kilogram (IPK). Currently, the NIST-4 watt balance is under construction and will be used to realize the unit of mass in the United States.

THE MAGNET SYSTEM DESIGN

The design of the NIST-4 magnet system was inspired by the BIPM watt balance group. Two opposing $\text{Sm}_2\text{Co}_{17}$ segmented rings are housed inside a 60 cm diameter 1021 steel yoke. The magnetic circuit guides the field such that an almost azimuthally uniform field permeates through a 3 cm wide, 10 cm tall air gap defined by the inner and outer yokes.

¹ Certain commercial equipment, instruments, or materials are identified in this paper in order to specify the experimental procedure adequately. Such identification is not intended to imply recommendation or endorsement by the National Institute of

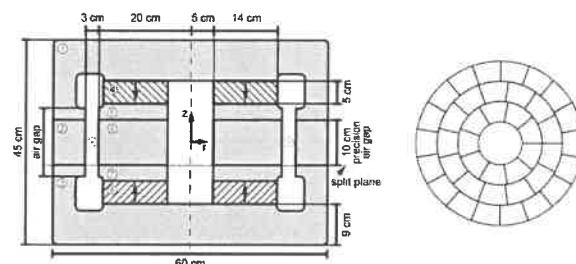


FIGURE 1. Left: a cross sectional view of the magnet system. The circles in the air gap represent the current in the coil and the arrows indicate the direction of magnetization. Right: top view of the segmentation of a $\text{Sm}_2\text{Co}_{17}$ disk.

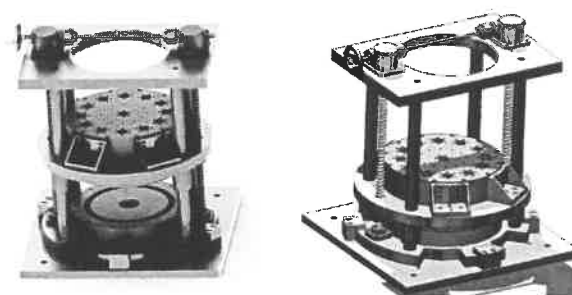


FIGURE 2. To insert the coil into the magnet, it is cranked open with a splitter device. Left: the top two-thirds are removed from the bottom third, exposing the internal air gap. Right: magnet in the closed position.

While NIST was responsible for the conceptual design of the magnet system, the manufacturing was contracted to Electron Energy Corporation¹ (EEC) with a specified field strength of 0.5 Tesla and a "flat profile" uniformity of $\pm 0.01\%$, or $\Delta B_r/B_r < 2 \times 10^{-4}$. $\text{Sm}_2\text{Co}_{17}$ is used for the active magnetic material. In order to magnetize the $\text{Sm}_2\text{Co}_{17}$, each ring is broken up in 40 sectors. Each 1 kg

Standards and Technology, nor is it intended to imply that the materials or equipment identified are necessarily the best available for the purpose

segment was sintered and magnetized by a capacitor bank individually.

For each of the 80 segments, the flux characteristics were measured and recorded. The position of every segment was then carefully chosen such that the resulting azimuthal and orthogonal fields were as similar as possible. Post-assembly measurements indicated the two magnet disks were within 0.2 % of each other.



FIGURE 3. Exploded view of the magnet system. The 1021 steel yokes are shaded in light grey and the Sm2Co17 disks are comprised of 40 individually magnetized wedges. The centering tubes are made from stainless steel and their purpose is to keep all the internal components concentric. Dowel pins maintain outer yoke alignment and steel screws lock the assembly together. Stainless steel retaining rings constrain the magnet wedges in the radial direction.

To facilitate the assembly process and to control the repulsive forces between segments, vacuum compatible epoxy was used to secure the magnet segments to the inner yoke plate while a steel retaining ring constricted them radially. A precision ground inner and outer yoke are bound by two identical outer yoke lids, providing the closed magnetic field circuit. Twelve holes on the top and bottom face allow access for hanging an inductance coil inside the precision air gap.

In order to insert the coil, a magnet splitting device was designed. Such a contraption must be able to constrain the unstable state once the magnet has been split and maintain vertical alignment. Each of the four 5 cm diameter steel guide rods is coupled with a frelon-coated linear bearing. The top and bottom yokes are bolted to the magnet splitter frame. Cranking a synchronized set of twin 1/64 gear reducers slowly rotates two lead screws, gradually

separating the magnet and providing access to the internal air gap.

METHODS OF CHARACTERIZATION

Two methods were implemented in searching for an unwanted vertical magnetic flux gradient in the air gap: a hall probe and a gradiometer coil (GMC). Both EEC and NIST conducted initial independent hall probe measurements by scanning an 80 mm vertical sweep through each of the 12 access holes (Lakeshore MMZ-2518-UH and HMMT-6704-VR for the EEC and NIST system, respectively). EEC conducted similar hall probe measurements on two different days spaced 1.5 weeks apart (Fig. 4). They measured each access hole at 7 different heights inside the air gap. We measured each access hole continuously by attaching the hall probe to a motorized vertical translation stage.

The gradiometer coil, mean diameter of 433 mm, consisted of two concentric coils wound in series opposition. They are displaced vertically but wound on a single former. A motor driven translation stage moved the coil through the air gap at a constant 2 mm/s. One voltmeter measured the induced voltage of one coil, another the difference, which was used to calculate the gradient of the radial flux density along the vertical.

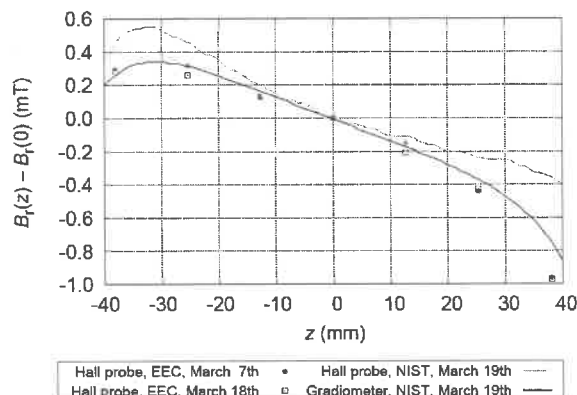


FIGURE 4. EEC and NIST hall probe/gradiometer coil measurements after the initial assembly. All four measurements indicate an approximate -13 $\mu\text{T/mm}$ gradient.

All four measurements were in good agreement, with their mean values differing by about 4 mT (< 8 parts per thousand) (Fig. 4). Each data point is comprised of averaging all 12 holes at the specified z location. The field slopes are all

approximately $-13 \mu\text{T/mm}$, about a factor of 10 larger than the desired field uniformity. These initial measurements also indicate a variation in the radial flux density of at least 1 mT, failing the requirement of $\Delta B_r/B_r < 2 \times 10^{-4}$. Hence, field profile tuning adjustments were necessary.

FIELD PROFILE ADJUSTMENT

Two methods were attempted to fine-tune the field profile: (1) regrinding the outer middle yoke to make a 0.05 degree taper of the air gap geometry, and (2) introduce a tilt separation between the top two-thirds and bottom third of the magnet to increase the reluctance in the bottom yoke.

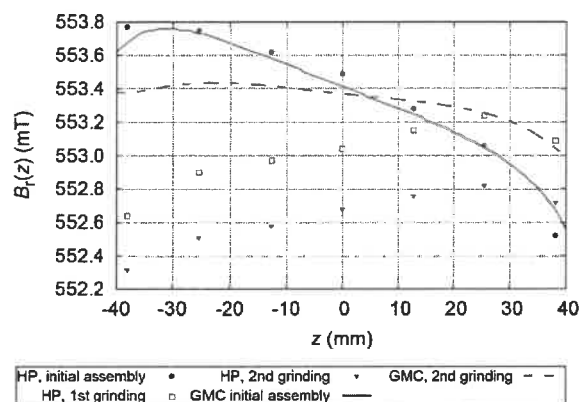


FIGURE 5. These data points represent the magnetic field measured with the hall probe and gradiometer coil before and after regrinding.

REDGRINDING THE OUTER YOKE

The method of regrinding the outer middle yoke (part number 2 in Fig. 1) was to add a slight taper to the air gap such that the top gap is nominally 3.000 cm wide and the bottom is 3.008 cm wide. After regrinding, EEC measured again the field profile to discover the grinding process overshot by 50%, resulting in a slope change from $-13 \mu\text{T/mm}$ to $+7 \mu\text{T/mm}$. The outer middle yoke was sent back for a second regrinding, this time with the instruction to grind the gap with a less aggressive taper, 3.003 cm at the top and 3.008 cm at the bottom. This procedure failed to indicate any change; the first grinding and the second grinding showed the same gradient. Fig. 5 shows the results of the initial assembly, the first grinding and the second grinding.

From this, we decided to abandon the hall probe measurement setup. It is likely that the translation of the hall probe(s) were not purely vertical and/or nonlinear. For example, if the measurement setup was to blame, the "vertical" probe trajectory would only need a parasitic horizontal component of 0.24 mm over a path length of 80 mm to measure a $7 \mu\text{T/mm}$ gradient. While the probe was certainly positioned better than 1 mm in the center of the gap, an accuracy of 0.2 mm could not be ensured. After the second regrinding, another GMC measurement was taken, showing a $-3.5 \mu\text{T/mm}$ gradient [2].

SEPARATION METHOD

Since the grinding process proved ineffective, other options were explored. A separation method was implemented, i.e. a gap between the bottom third of the magnet and the top two-thirds was introduced. We found that a flat profile was achieved when the separation was approximately 0.5 mm high. A stable and uniform azimuthal air gap can be maintained by inserting aluminum shim stock pieces at several azimuthal locations.

This separation increases the reluctance in the lower yoke. Hence, the lower $\text{Sm}_2\text{Co}_{17}$ magnet contributes less flux to the radial field through the precision air gap. While this azimuthal air gap helped achieve the desired flat profile, it also introduced a physical connection to the outside environment, allowing flux to leak out of the magnet, compromising the shielding.

During this experimental process, we noticed that the field profile slope changed by a few $\mu\text{T/mm}$ every time the magnet was split and closed. The variability of the slope arose from non-parallel splitting, i.e. the magnet pieces initially tended to tilt before completely breaking contact. In this case, the lower part of the yoke touches the upper part on one localized spot along the edge. This redistributes the flux and concentrates it through the contact area, essentially walking the BH curve to the right and decreasing the relative permeability, μ . Even after the magnet is closed, it maintains a state of smaller relative permeability due to the hysteretic behavior of the BH curve. Hence, in the closed state the bottom yoke conducts the magnetic field worse and the flux in the gap is lowered [2].

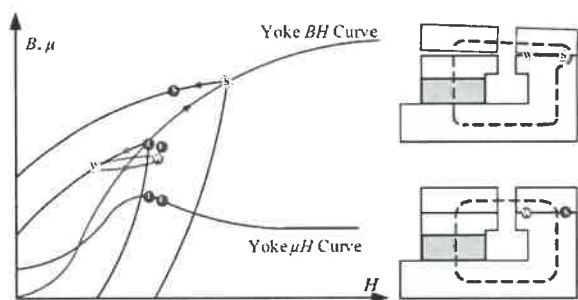


FIGURE 6. Schematic drawing showing the change in the outer part of the yoke before, during, and after opening with an angle. We consider the locations S and W at the small and wide side of the gap on the outer yoke. At the beginning, the magnetic state is given by the point I on the B - H curve. During opening, the point W moves along a minor hysteresis loop to smaller values of B and H . The point S moves along the major curve to a higher value of B and H . After closing, the point W moves almost back to the original point (the solid W). The point on the small side of the wedge moves along a minor loop to a new point (solid S) that has substantial more B . The mean value can be found at the point denoted by F for final point. Overall the μ_r of the yoke has decreased, reducing the flux in the air gap.

However, this new phenomenon proved beneficial for tuning the field and a procedure was developed. (1) The magnet is opened by a little more than 1mm. (2) A 0.5 mm thick shim piece with a size of approximately 5 cm by 5 cm is inserted in the 1mm gap at an azimuthal position α . (3) The magnet is closed. Due to the shim, the magnet closes in a tilted fashion and the iron at the azimuthal position $\alpha + 180^\circ$ is driven to the state with less relative permeability. Steps (1) through (3) are performed a total of six times, where the azimuthal position is advanced by 60° every time. After this, the iron remains at the less permeable state for the entire circumference. [1]

This shimming process is repeatable. We were able to reproduce the shimming procedure several times, yielding an almost identical field profile each time (Fig. 7).

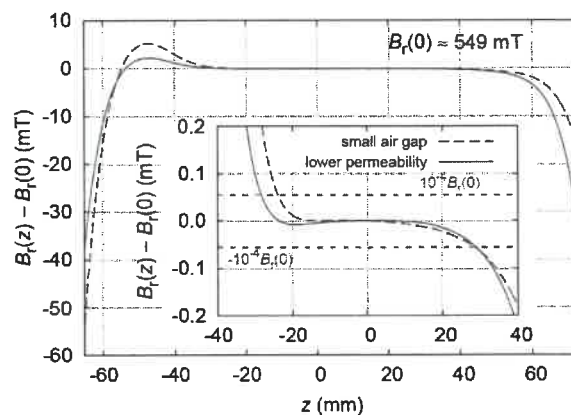


FIGURE 7. Radial flux density as a function of vertical position. The dashed line shows the effects of introducing a small tilt separation.

SHIMMING METHOD APPREHENSIONS

We have two concerns using this shimming method: How stable is the field obtained with this method? Does this process change the azimuthal symmetry of the field? We monitored the field profile for 3 days, taking data every 30 minutes and found the flux density drift in the center of the air gap to be about 2.5×10^{-10} T/h, or 0.5 parts per billion per hour. This is enough stability for a watt balance experiment where the flux integral is measured once per hour [2].

The azimuthal variation in the field symmetry proved harder to measure. This can only be done by measuring discrete points around the air gap, meaning the gradiometer coil would be obsolete since it measures the azimuthal integral. Using the hall probe, however, requires precise positioning. Due to the outer yoke design of the magnet, the air gap can be accessed through 12 discrete holes, the middle region of the air gap being 22.5 cm from the top surface. The hall probe must not only be able to reach this location but also be reproducibly inserted in all 12 holes. For example, a 1 mm deviation in the middle of the gap results in a 2.3 mT change. The measurements before and after magnet shimming indicated a similar maximum difference of 1.5 mT. Thus, it is unknown whether or not the uncertainty arises from field inhomogeneity or hall probe positioning error.

CONCLUSION

Both shimming methods have allowed us to achieve the desired field profile uniformity of .01%

(50 uT) over a 50 mm tall section of the air gap. In the end, we chose to shape the field with the method of alternating opening/closing tilt shimming because of higher repeatability and maintaining a closed magnet system.

The magnet has recently been installed in our watt balance. Most of the mechanics of the new watt balance have also been mounted on top of the magnet, inside the vacuum chamber. A handful of velocity mode (the calibration mode of a watt balance experiment to characterize the magnetic field) measurements have already been conducted but no further investigation has been done for uniformity changes. Since uniformity is a larger factor in the weighing mode, we will revisit this topic when we begin taking weighing mode measurements.

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FABRICATION OF THE MAGNETIC CIRCUIT FOR THE BIPM WATT BALANCE

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INTRODUCTION

Since 1889 the International Prototype of the Kilogram, kept at the BIPM, had served as the definition of the unit of mass in what is now known as the International System of Units (SI). It is the last material artefact to define a base unit of the SI, and it influences several other base units. This situation is no longer adequate in a time of ever increasing measurement precision.

It is therefore planned to redefine the unit of mass by fixing the numerical value of the Planck constant [1,2], similar to the present definition of the metre, which is based on a fixed numerical value of the speed of light. To ensure that the redefinition does not introduce a discontinuity in the magnitude of the kilogram, as a first step, a highly accurate determination of the Planck constant in the present SI system needs to be carried out, with a relative uncertainty in the order of 1 part in 10^8 .

The most promising experiments for this purpose are the watt balance [3] and the X-ray crystal density technique [4]. A watt balance is an electromechanical apparatus which compares mechanical and electrical power and makes use of two macroscopic quantum effects, the Josephson effect and the quantum Hall effect, thus creating a relationship between a macroscopic mass and the Planck constant. The BIPM is developing a watt balance to ensure that in future the unit kilogram can be derived from the Planck constant.

In the BIPM watt balance a current-carrying coil is placed in the circular air gap of a magnetic circuit (FIGURE 1). The sources of magnetic flux are two disks of $\text{Sm}_2\text{Co}_{17}$ magnets (number 9 in FIGURE 2), with opposite direction of magnetization. A closed yoke (numbers 1, 2, 3 and 4 in FIGURE 2) made of a high permeability Fe-Ni alloy (Supra 50, ArcelorMittal) is used to concentrate the flux and to shape the radial magnetic field within the air gap, where the coil

will be placed. The mean diameter of the air gap is 250 mm, its width is 13 mm and the height is 80 mm. The flux density in the centre of the air gap is 0.5 T.

An important requirement for the watt balance operation is that the flux density in the air gap is highly constant in the vertical direction, which requires that the width of the air gap is uniform to within several micrometers. In addition, the outer and inner poles shall be centered to better than 50 micrometers. This requires machining tolerances at the level of 1 micrometer and the use of a dedicated assembly device, capable of accurately positioning the parts in the presence of very large magnetic forces, up to 10 kN.

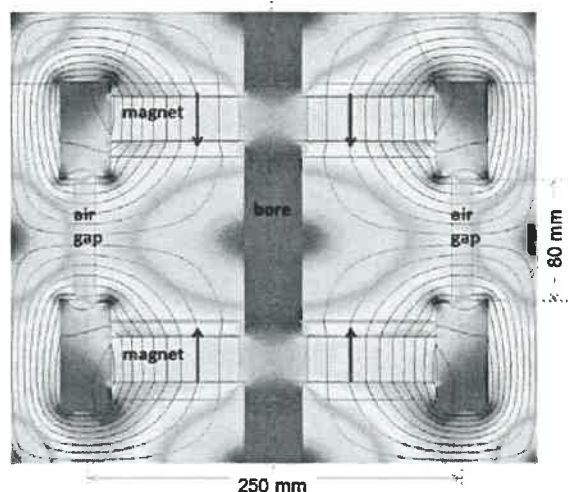


FIGURE 1. Magnetic circuit for the BIPM watt balance with magnetic flux lines. The system is rotationally symmetric around the vertical axis.

From the manufacturing point of view this results in several critical parts determining the overall accuracy of the assembly and thus the feasibility for the experiment. The high permeability Fe-Ni alloy furthermore challenges the machining equipment since experience with this material is

rare. The housing (number 1 in *FIGURE 2*), a bucket-like part about 350 mm in diameter and 300 mm in height, contains cylindrical and planar surfaces on the inside and outside with specified flatness and cylindricity as well as concentricity of a few micrometers. The core (number 2 in *FIGURE 2*) is a cylinder of Supra 50 iron-nickel soft magnetic alloy, 110 mm tall and 240 mm in diameter. It has to be cylindrical to better than 1 micrometer with perpendicular and planar top and bottom surface. The cover segments (numbers 3 and 4 in *FIGURE 2*) additionally contain angular surfaces specified to the same accuracy level.

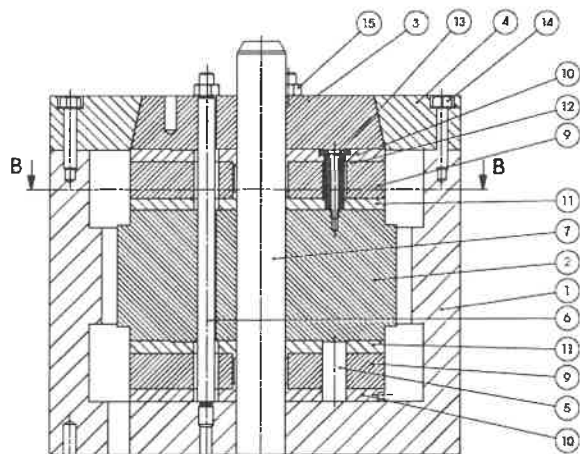


FIGURE 2. Cross section of the BIPM watt balance magnet.

PRECISION MANUFACTURE

The critical components of the watt balance are made of Supra 50, a magnetically soft iron-nickel alloy with a density similar to steel but a coefficient of thermal expansion 30-40% less than typical carbon steel. The two components forming the nominally Ø250 mm circular air gap must have micrometer-level cylindricity and the larger of the two has a mass of 90.8 kg. To achieve this level of precision in such heavy parts, manufacturing skills, tools, and techniques were leveraged that were developed over 50 years of air bearing spindle manufacture. Instrumental to that effort is the retrofit of all of the grinding machines at Professional Instruments with compact ultra-precision air and oil hydrostatic spindles. Furthermore, on-machine gaging enables process monitoring and correction without removing the part from the machine. A bespoke indicating micrometer features a diamond lower anvil held in contact by a parallel flexure that also maintains vertical alignment of the metrology frame. A CEJ

Mikrokator with a diamond tip measures displacement with 0.25 μm resolution. However, this measurement alone is not sufficient to certify cylindricity and perpendicularity. Independent verification of cylindricity required measurement of roundness, straightness, and perpendicularity using a roundness tester built upon a Moore No. 3 Universal Measuring Machine. Final diameter measurements were determined with a Zeiss Contura G2 CMM in a controlled environment while monitoring and correcting for thermal expansion. The estimated cylindricity from the combination of these measurements was 0.6 μm for the Core OD and 1.4 μm for the Housing ID.

QUALIFICATION OF SINGLE COMPONENTS

Successful application of the watt balance magnet requires for highly accurate parts to allow for high accuracy of the assembled system. Specified form, shape and dimensional tolerances in the range of the nominal accuracy of conventional measuring equipment at parts of up to 90 kg require for special measuring procedures. Despite on-machine gaging and documented dimensional accuracy, independent measuring of the shape tolerances with different equipment was performed to enhance reliability of the measuring results.

Dimensional tolerances are less strict and were controlled using a CMM. Checking shape accuracy the second time was done with an adapted UP-machine tool. Work piece weight, shape and size did not allow application of standard measuring equipment. Even though ultra-precision machine tools are known to provide high accuracy linear and rotary axes and motion, the real accuracy was verified directly before the measurements. Linearity and parallelism as well as perpendicularity of the axes x, z and c were checked (*FIGURE 3*).

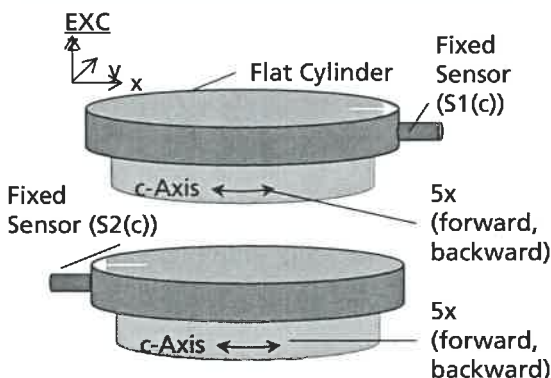
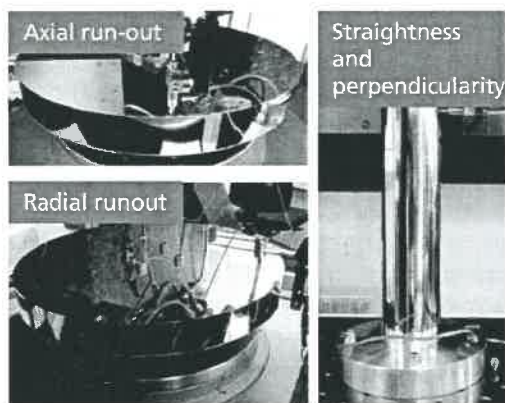


FIGURE 3. Verification of feasibility of ultra-precision machine tool for detection of form tolerance.

Accuracy of the single linear and rotary axes as well as their alignment was verified using capacitive sensors in combination with ultra-precision machined references. To eliminate influence of topology errors of the references reversal method and back-and-forth measurement were applied.

Subsequently, cylindricity, flatness and perpendicularity of the single parts were measured using capacitive sensors with a setup comparable to conventional roundness testers, see FIGURE 4. Cylindricity was calculated from several roundness measurements along the cylindrical surface. Measuring the bottom surface in the same setup allowed for the calculation of flatness and perpendicularity. The measured results were in the range of the ones stated above and the good dimensional accuracy and form accuracy was verified.

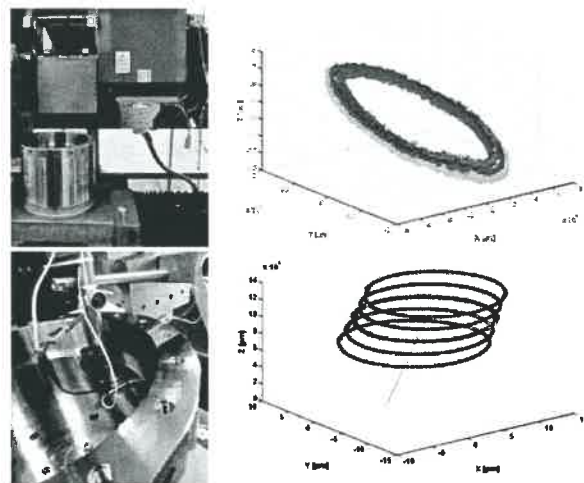


FIGURE 4. Measuring of housing: flatness (top), cylindricity (right).

QUALIFICATION OF THE ASSEMBLY

Besides accuracy of the main parts, accuracy of several other parts as well as the assembly process influence the accuracy of assembly. Gap width in axial direction had to be constant to below $3\text{ }\mu\text{m}$ in a depth of 65 to 145 mm. Placement of core to housing is most critical since this determines constancy of gap width. The core is not placed directly on the lower magnet disk (parts 9, 10, 11 in FIGURE 2) but is supported by three titanium columns (one visible on FIGURE 2 as part 5) of nominally equal length. Thus, especially gap width was verified after a test assembly, without the magnets in place, before final assembly of the complete system.

For verification measurements, fiber-optic systems, flat eddy current sensors and capacitive sensors were discussed. Parallel detection of the inner and the outer surface of the gap instead of successively measuring and subsequently calculating the deviations was identified to be beneficial. Influence of straightness inaccuracies of the linear axes, positioning inaccuracies and vibrations as well as thermal influences could thus be reduced. A schematic of the measuring setup is given in FIGURE 5. Besides the length of the sensor, especially a slim design of the sensor head had to be realized.

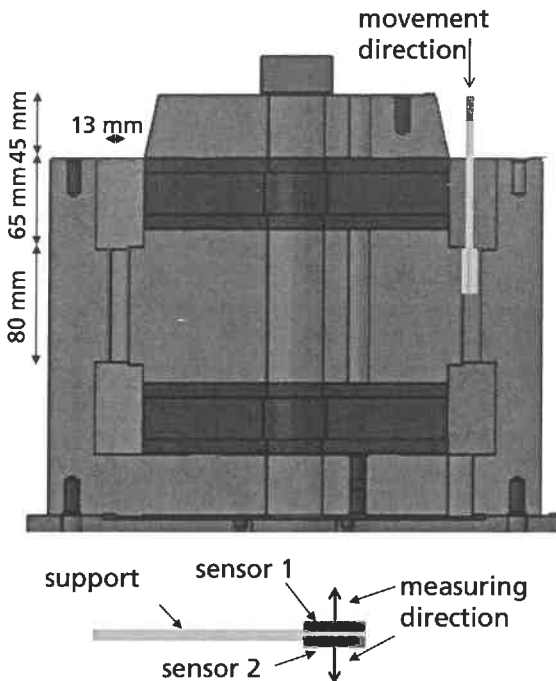


FIGURE 5. Alignment of capacitive sensors for detection of gap width between inner and outer magnetic poles.

Two capacitive sensors facing opposite directions with a nominal resolution of < 80 nm were selected. They offered the best compromise between achievable accuracy, size and cost. Measuring against the curved surfaces of housing and core did not harm accuracy since $D \gg d$ (D : diameter of core; d : measuring spot). A trial assembly, replacing the magnets with un-magnetic placeholders could thus be carried out. At every 30° the gap was controlled three times to ensure unbiased measurements (example see FIGURE 6). The sensor was moved with a motorized translation stage at constant velocity. The vertical variations in the central part of the air gap were well below 1 micrometer.

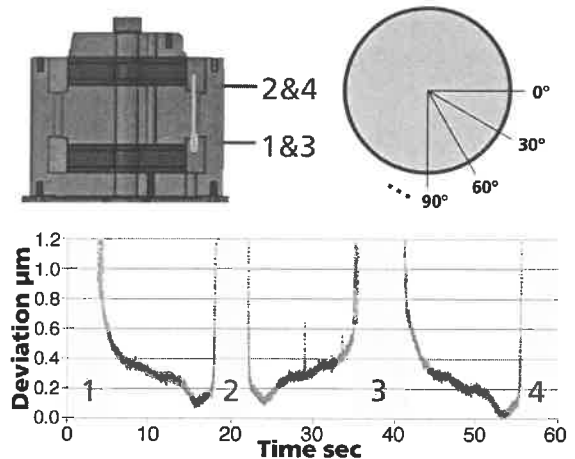


FIGURE 6. Gap measurement: setup (above) and example measuring result (below)

ASSEMBLY OF THE MAGNETIC CIRCUIT

A special assembly device had been developed to accurately position the parts. A particular difficulty was the strong magnetic attraction of up to 10 kN between the magnets and the soft iron parts, which required a very stiff structure. FIGURE 7 shows the process of assembling the core with one of the magnet disks. The magnet disk (grey) is solidly fixed to an aluminum plate. The core (dark green) is placed on three titanium rods (blue) which are gently moved downwards by a manually operated gear system below the aluminum plate. A central, closely fitting 40 mm diameter titanium column (green) served in guiding the parts. After contact, the parts stick together due to the magnetic attraction.

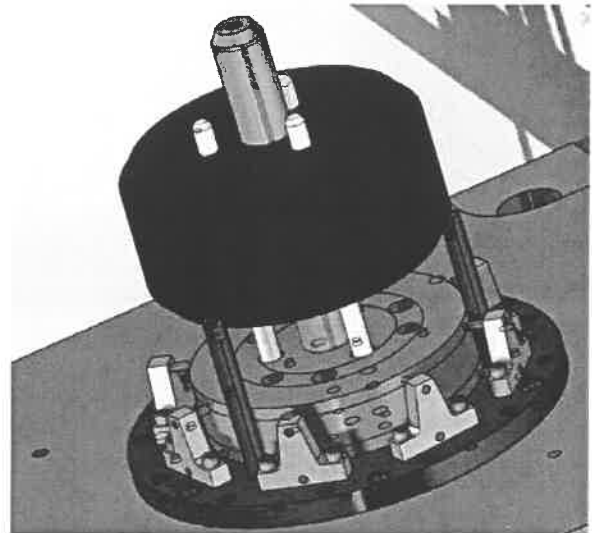


FIGURE 7. Assembly of the core (dark green) with one of the two magnet disks (grey).

FIGURE 8 shows the insertion of the core and the two magnet disks into the bucket-like housing.

A particular difficulty consisted in finding the right compromise for the diameter of the central titanium alignment column: not too small to guarantee good centering, and not too large to avoid blocking of the parts. A nominal gap of 10 μm was finally chosen and the column was lubricated with graphite.

After the assembly, the geometry of the air gap was verified with the double capacitive sensor described above. It was found that at different angular positions, the gap width was uniform to within 2 μm over the central 30 mm. The poles were centered to within 10 μm . Both are within specifications and the magnet is now inserted into the BIPM watt balance apparatus.



FIGURE 8. Insertion of the inner parts of the magnetic circuit into the housing.

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DESIGN AND PERFORMANCE OVERVIEW OF ADVANCED LIGO

TWO-STAGE VIBRATION ISOLATION AND ALIGNMENT SYSTEM

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INTRODUCTION

This paper gives an overview of the vibration isolation and positioning platform used in Advanced LIGO to support the interferometer's core optics [1-2]. The design of this system is based on the architecture of the platform used during the prototyping phase [3]. This five-ton two-and-half-meter wide system operates in ultra-high vacuum. It features two stages of isolation mounted in series. The stages are imbricated to reduce the overall height. Each stage provides isolation in all directions of translation and rotation. The system is instrumented with a unique combination of low noise relative and inertial sensors. The active control provides isolation from 0.1 Hz to 30 Hz. At the LIGO sites, it brings the platform motion down to $10^{-11} m/\sqrt{Hz}$ at 1 Hz. Active and passive isolation combine to bring the platform motion below $10^{-12} m/\sqrt{Hz}$ at 10 Hz. The passive isolation lowers the motion below $10^{-13} m/\sqrt{Hz}$ at 100 Hz. The next sections give an overview of the system's architecture, control strategy, design features and isolation results. A detailed presentation of all those aspects is given in [4-5].

SYSTEM OVERVIEW

A conceptual representation of the two-stage system is shown in Fig. 1. It represents the structure and the spring components in a schematic section view. Actuators and sensors are not displayed. The system is made of three main sub-assemblies: a base called "Stage 0" and two suspended stages called "Stage 1" and "Stage 2". The stages are imbricated to minimize the volume occupied. The bottom plate of Stage 2 is the optical table on which the Advanced LIGO equipment is mounted. Stage 1 is suspended from Stage 0, and Stage 2 is suspended from Stage 1. The spring assemblies provide horizontal and vertical flexibility. They are symbolically represented by helicoids. The springs decouple the stages from each other in all directions of translation (called Longitudinal, Transverse and Vertical in Fig. 1) and all directions of rotation (called Pitch, Roll and Yaw in Fig. 1). The system is designed to minimize the cross couplings between degrees of freedom. In each direction, the system behaves as a two-mass-spring system.

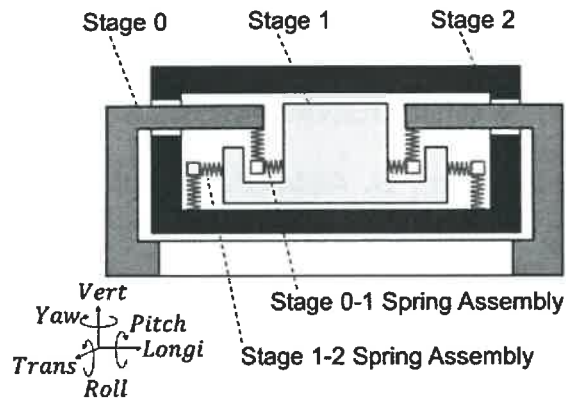


FIGURE 1. Multi-stage isolation concept

CONTROL STRATEGY OVERVIEW

Stage 1 and Stage 2 are controlled actively. Each of the twelve degrees of freedom are controlled independently. Fig. 2 shows the feedback control block diagram for a single degree of freedom, where X_n is the degree of freedom under control. The subscript value n can be 1 for Stage 1, or 2 for Stage 2. The stage motion X_n is disturbed by the input motion X_{n-1} through the seismic path called P_s . It is controlled with the force F_x through the force path called P_F .

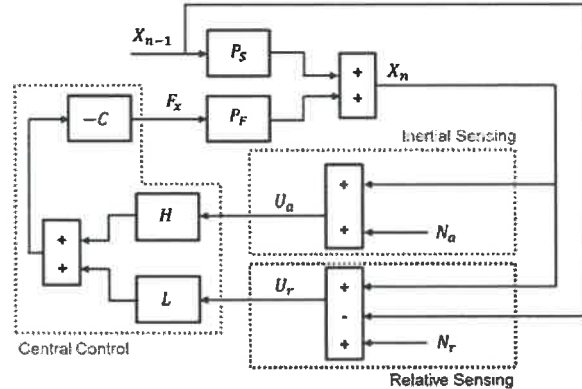


FIGURE 2. Control block diagram

On each of the two active stages (Stage 1 and Stage 2), two types of sensors are used: inertial and relative sensors. The inertial sensors (geophones or seismometers) are used to measure the stage's absolute motion (X_n) which is necessary to provide active seismic isolation (isolation/decoupling from the previous stage). The absolute measurement (inertial instrument signal) is called U_a in Fig. 2. It includes a noise component called N_a . This noise is dominant at low frequency where the inertial sensor loses sensitivity. The relative sensors are capacitive

gauges measuring the differential motion between the stages ($X_n - X_{n-1}$). They are used for DC and low-frequency positioning. The relative measurement is called U_r . It includes a noise component called N_r . This noise is much lower than the inertial sensor noise at low frequencies. A sensor fusion is used to combine the absolute and the relative measurement. The inertial sensor signal is filtered with the high-pass called H . The relative sensor is filtered with the low-pass L . The output of these filters are summed to form the error signal. A compensator C is used to command the control force F_x .

The components of this block diagram are summarized in Eq. (1) to (4), assuming that the absolute measurement (U_a) and relative measurement (U_r) are calibrated in displacements units. This is done practically using digital filters inverting the instruments frequency response. Equation (1) gives the stage motion as a function of the input motion (disturbance) and the control force. Equation (2) gives the control force as a function of inertial measurement (U_a) and the relative measurement (U_r). Equations (3) and (4) introduce the sensor noise. To simplify the control design, the low-pass and high-pass filters are designed to be complementary [6], as shown in Eq. (5). Equation (6) gives the closed loop power spectral density (X_n^2) assuming the input motion and the sensor noises are uncorrelated. The first term shows the contribution of the input stage motion (X_{n-1}). The second term shows the contribution of the absolute (inertial) motion sensor noise (N_a). The third term shows the contribution of the relative motion sensor noise (N_r). Assuming large loop gain in the control bandwidth, the amplitude spectral density can be written as shown in Eq. (7). The input motion contribution is filtered by the low-pass filter L . Therefore, the lower the cutoff frequency the better for the isolation, but the high pass-filter H and low-pass filter L must also be adequately designed to minimize the sensor noise contribution. The optimization consists of designing complementary filters that provide both adequate isolation and filtering of the instrument's noise.

$$X_n = P_s X_{n-1} + P_F F_x \quad (1)$$

$$F_x = -C (H U_a + L U_r) \quad (2)$$

$$U_a = X_n + N_a \quad (3)$$

$$U_r = (X_n - X_{n-1}) + N_r \quad (4)$$

$$L + H = 1 \quad (5)$$

$$X_n^2 = \left(\frac{P_s + L C P_F}{1 + C P_F} \right)^2 X_{n-1}^2 + \left(\frac{H C P_F}{1 + C P_F} \right)^2 N_a^2 + \left(\frac{L C P_F}{1 + C P_F} \right)^2 N_r^2 \quad (6)$$

$$\lim_{(C P_F) \rightarrow \infty} X_n = \sqrt{(L X_{n-1})^2 + (H N_a)^2 + (L N_r)^2} \quad (7)$$

DESIGN OVERVIEW

The base of the system (Stage 0), the first suspended stage (Stage 1) and the second suspended stage (Stage 2) are shown in grey shades in the Computer Aided Design (CAD) representation in Fig. 3. The picture shows how Stage 1 and Stage 2 are imbricated to reduce the system's volume.

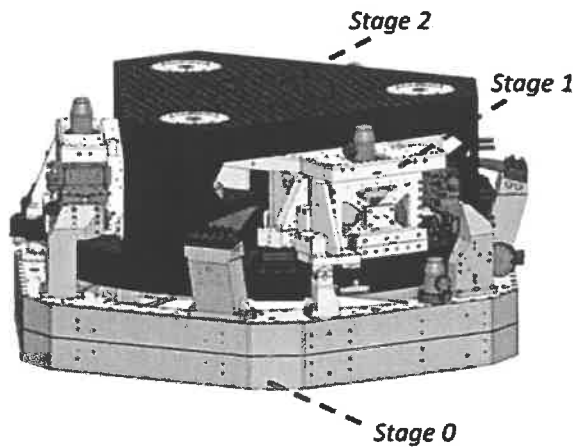


FIGURE 3. Two-stage isolator

Stage 1 is suspended from Stage 0 using three sets of blades and flexures. Stage 2 is suspended from Stage 1 using three sets of blades and flexures similar to those used between Stage 0 and Stage 1. The blades are designed to provide the vertical flexibility, and

the flexure rods are designed to provide the horizontal flexibility. A cross section of the Stage 0-1 spring assembly is shown in the CAD representation in Fig. 4.

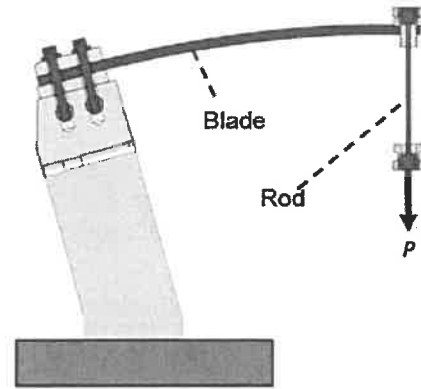


FIGURE 4. Flexure components

The two stages' mass, inertia properties, and the spring's stiffness are chosen to obtain suitable rigid-body natural frequencies. They must be low enough to provide adequate passive isolation, but the springs must not be too compliant so that the system remains easy to commission and operate. LIGO experience acquired during prototyping of vibration isolation systems showed that rigid-body natural frequencies in the 1 Hz to 7 Hz range provide an excellent compromise.

To reduce the complexity of the control strategy and to facilitate operation, the two-stage system has been engineered to minimize the couplings between the degrees of freedom in the Cartesian basis. The spring components have been designed to obtain in-phase rigid-body modes in the 1 Hz to 2 Hz range (the two masses moving in phase at the resonance), and out-of-phase rigid-body modes in the 5 Hz to 7 Hz range (the two masses moving out of phase at the resonance).

For the active control, six coarse (high force) electromagnetic actuators are used between the base and Stage 1. Six smaller fine (low force) electromagnetic actuators, are used to actuate between Stage 1 and Stage 2. Stage 1 is instrumented with three different sets of sensors:

- 6 MicroSense (formerly ADE technologies) capacitive position sensors between the base and Stage 1, called "coarse". They are used for low frequency relative positioning.
- 3 Trillium T240 seismometers: these 3-axis seismometers provide the low frequency inertial

measurements necessary to provide active seismic isolation.

- 6 Sercel L-4C geophones: these single-axis seismometers provide the high frequency inertial sensing necessary to provide active seismic isolation at higher frequencies.

Stage 2 is instrumented with two different sets of sensors:

- 6 MicroSense capacitive position sensors between Stage 1 and Stage 2, called "fine". They provide the low frequency relative positioning.

- 6 Geotech GS-13 seismometers: these single-axis seismometers provide the inertial sensing needed for active seismic isolation.

The horizontal actuators are positioned with respect to the spring assembly and the stage center of mass to minimize the cross couplings between horizontal and tilt motion. The magnets are positioned in pairs of North-South and South-North dipoles to reduce the magnetic field which escapes the actuator assembly. The actuator has a return yoke to minimize the escaped magnetic field which can interfere with the sensitive equipment surrounding the platform. Relative sensors and actuators are collocated as shown in Fig. 5.

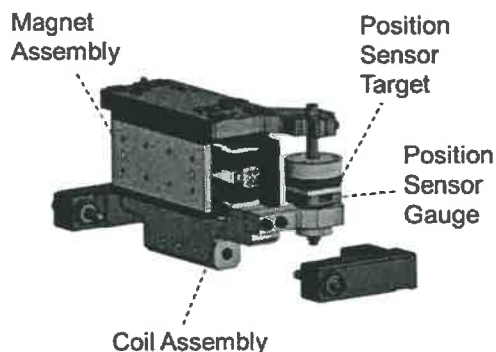


FIGURE 5. Actuator and relative sensor

In order to be used in ultra-high vacuum, each seismometer is mounted in a sealed chamber, as shown in Fig. 6. The chamber is custom-made for this application. It is made of Stainless Steel 304. The welded parts are subjected to a thorough leak-check process to validate the fabrication. For this test, the chamber is filled with Helium, and subjected to a RGA scan. For the final assembly, the pod containing the instrument is filled with Neon, which is used as a tracker for leak detection during operations. Additionally, each pod is instrumented with

pressure sensors to help identify a faulty pod in case of a leak.

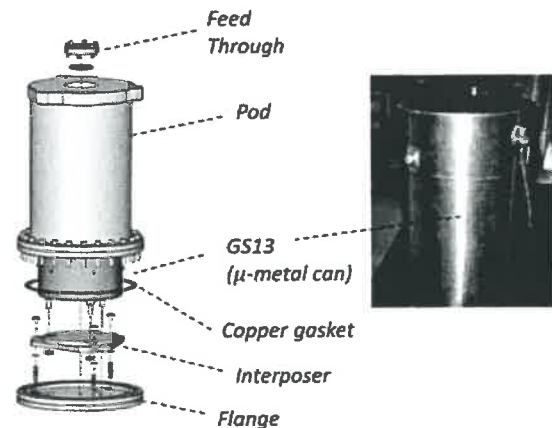


FIGURE 6. Sealed inertial instruments

A unit undergoing the test process at the LIGO Hanford observatory is shown in Fig 7. In this picture, 1100 kg of dummy mass is mounted on the top plate to float the stages during the testing phase. Once the testing is completed, the dummy mass removed, the unit is moved to the detector area where interferometer components are attached to the inverted (down-facing) optical table of Stage 2.

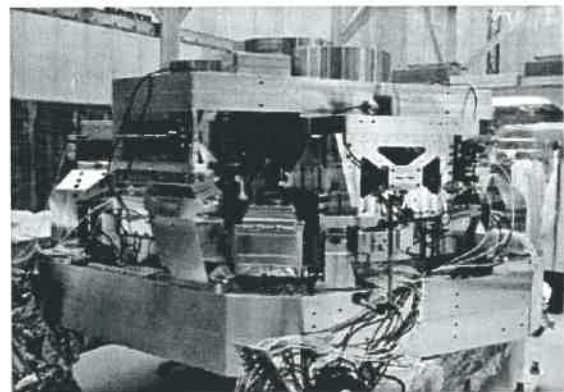


FIGURE 7. Two-stage isolator in testing

PERFORMANCE

The curves in Fig. 8 show transfer functions from a force (or torque) applied on Stage 1 to the motion of Stage 2. Sensors and actuators are combined to drive and sense the degrees of freedom in the Cartesian basis. Active inertial damping is used damp the rigid-body mode resonances. The transfer functions are normalized by the Stage 0-1 spring stiffness so that the DC response is equal to unity.

The plot shows the responses in the longitudinal and yaw directions. The dashed curve shows

the transfer function from a torque applied along the yaw axis, to the rotation motion around the same axis. The solid curve shows the transfer function from a force applied along the longitudinal axis, to the translation motion along the same axis. These measurements show that the slope above the natural frequencies is function of the fourth power of frequency. Both curves are under -40 dB of magnitude at 10 Hz, and under -120 dB of magnitude at 100 Hz. Similar results are obtained for all degrees of freedom.

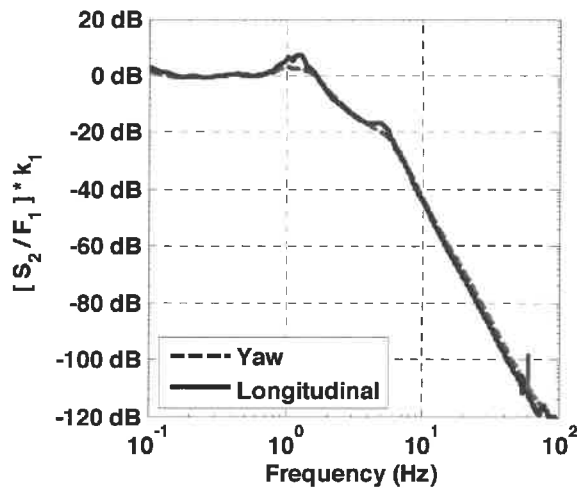


FIGURE 8. Force Driven Transfer Function

Fig. 9 shows transmissibility measurements. All of the degrees of freedom are under control as described in the control section. Hydraulic actuators are used to drive Stage 0 motion. Geophones mounted on the structure supporting Stage 0 are combined to estimate the input motion in the Cartesian basis. The inertial sensors on Stage 2 are used to estimate the output motion along the direction of the drive. Transmissibility measurements are performed in all directions of translation and rotation. Fig. 9 shows the transmissibility up to 15 Hz. Above those frequencies the frame supporting Stage 0 deforms. Consequently, the sensors mounted on this frame do not provide an accurate measurement of the input motion.

The dashed curve shows transmissibility from Stage 0 to Stage 2 in the yaw direction. The controllers are tuned to provide approximately 20 dB of isolation at 1 Hz. Further isolation can be obtained for this degree of freedom but it is often not necessary (ground yaw motion is typically small, and sensor the signal is often close to sensor noise).

The solid curve shows transmissibility from Stage 0 to Stage 2 in the longitudinal direction. The low frequency performance achievable in this direction (and in transversal) is limited by tilt horizontal coupling. At low frequency (around 100 mHz and below), the signal is dominated by tilt rather than horizontal motion [7-8]. In this example, the filters are tuned to provide 55 dB of attenuation at 1 Hz.

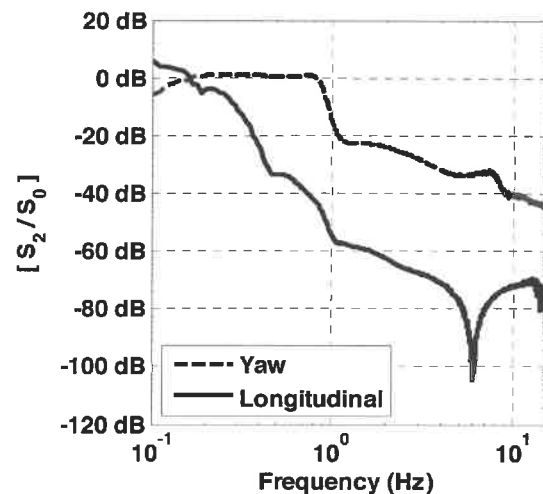


FIGURE 9. Transmissibility

Fig 10 presents the platform's absolute motion performance. A Streickeisen STS-2 is used to estimate the translational absolute ground motion in the longitudinal direction. The inertial sensors on Stage 2 are used to estimate the rigid body motion of the platform's output in the same direction.

The ground motion is shown by the solid line, the Advanced LIGO requirements are shown by the dash-dotted line, the inertial sensor theoretical noise is shown by the dotted line, the platform's motion measurement is shown in by the dashed line.

Up to 20 Hz, the platform motion is at or below the requirements. Above 20 Hz, the platform motion is very close to requirements. The small mismatch with requirements is inconsequential since seismic motion will not dominate the interferometer noise at those frequencies (the initial requirements included sufficient margin for such mismatch).

In the mid-band frequency [0.5 Hz to 10 Hz], the measurement is at or under the sensor noise. The portion of the curve under the sensor noise over-estimates the actual performance since those sensors are in loop. An out of loop witness sensor would be necessary to evaluate

accurately the absolute motion in the frequency band. In-loop measurements under the theoretical sensor noise, however, indicate that there is room to sustain larger input motion and still maintain similar isolation performance. In this measurement done during the summer time at Hanford, the input motion was near $10^{-9} \text{ m}/\sqrt{\text{Hz}}$ at 1 Hz. Measurements show that the motion at Livingston during the winter time can be more than 10 times larger. For such input, the output motion would still be near or slightly above the sensor noise. These results indicate that the two-stage system should operate at or near requirements at most times.

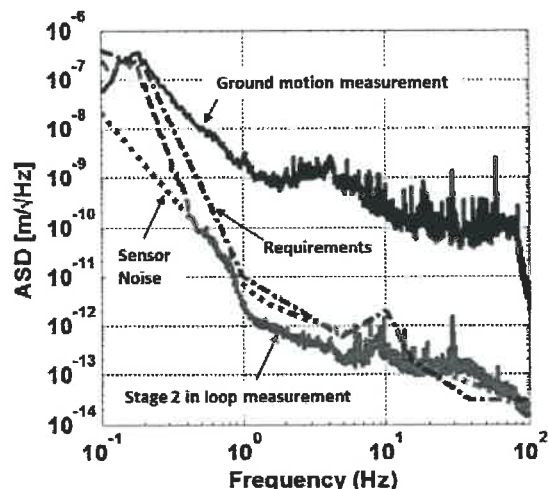


FIGURE 10. Absolute Motion Measurements

CONCLUSION

An extensive engineering effort has been led during the past several years to develop the final version of the two-stage vibration isolation system for the Advanced LIGO observatories. The goal was to engineer a system not only to meet very high performance criteria, but also suitable for timely production, assembly, testing and commissioning for a series of 15 units. This paper gave an overview of the system's design, control strategy and isolation results.

During the past two years, 13 units have been assembled and tested. The last two units are under construction and will be completed by the end of 2014. All the units tested show extremely reproducible results and characteristics. Five units are currently in use at each of the LIGO observatories and performing at the very high level of isolation required for Advanced LIGO. Detailed information regarding the design, prototyping, and production of these platforms can be found in [4-5].

ACKNOWLEDGMENTS

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A MULTI-DOF ACTIVE VIBRATION ISOLATION SETUP FOR A CORIOLIS MASS FLOW RATE METER

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INTRODUCTION

A Coriolis mass flow meter (CMFM) for small mass flows, as presented by Mehendale [1,2], is limited from measuring more accurately due to vibration disturbances introduced through the device's frame as shown by Van de Ridder [3]. Considerable improvement in measurement performance is expected from the application of passive and active vibration isolation strategies. Using feedback control, an attenuation of 42 dB of translational disturbances is expected [4]. Where the field quantity definition of dB used, i.e. $\text{value}_{\text{dB}} = 20 \log_{10}(\text{ratio})$. The component of the CMFM that needs to be isolated from disturbances is the tube window (see figure 1). This fluid-conveying tube is actuated in oscillation around the actuation-axis. A fluid flow in this oscillatory rotating tube results in a Coriolis force induced motion around the Coriolis-axis in the order of sub nanometres for the lowest flow range. A mixed signal of the actuation- and Coriolis-deflection is measured at the sensing locations, where the phase difference of both sensor measurements is linearly dependent on the mass flow, as derived by Mehendale [1].

This work extends the mentioned research with the design, modelling and validation of an active vibration isolation system that serves as a proof of principle for multiple vibration isolation strategies. The designed system is experimentally validated and active vibration isolation by means of a feedback control scheme is tested. In this paper, the design of the system is discussed, followed by the properties obtained from a dynamic model. An attenuation of the main source of disturbance by 50 dB is expected theoretically. With implementation of the same feedback scheme, an attenuation of 48 dB is obtained experimentally. The work concludes with a discussion of the presented system and results.

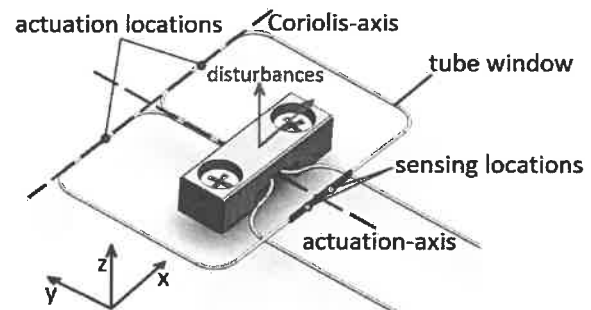


FIGURE 1. Solidworks® model of the fluid-conveying tube window and its socket.

CONCEPT DESIGN

Disturbances introduced through the device's frame can cause a measurement error. This occurs when the tube experiences motion similar to the Coriolis-deflection at the sensor locations. When this motion is around the measurement frequency of 170 Hz it will be indistinguishable from fluid flow and result in a measurement error. The two main contributors to this error are a translation perpendicular to the tube plane (z-direction) and a rotation around any axis parallel to the Coriolis-axis (x-axis) (see figure 1) as derived by Van de Ridder [5]. So the active vibration isolation system should isolate the tube window and its actuation and sensing components, together called the measurement stage, in both these directions from frame disturbances. Furthermore, the setup will be used to test if the tube window can be actuated by rotation of the tube's socket about the actuation-axis (y-axis),

Isolation from frame disturbances by means of feedback control is achieved in two steps. First, the measurement stage is suspended. The suspension provides passive vibration isolation above its resonance frequency (called suspension frequency in the following). The term suspension is used for a support with a designed stiffness significant to the involved dynamics. A suspension frequency of 30 Hz was determined by internal research as a lower

bound for sufficient mechanical properties when the device is unpowered and the control is deactivated. Second, a controller is designed in conjunction with sensors and actuators to close a feedback loop, providing active vibration isolation. This loop has the advantage that the suspension frequency can be lowered virtually, without compromising the low frequency behaviour of the system. Acceleration feedback is used with proportional and integral (PI) control action to add virtual mass and skyhook damping, respectively, to the system [6]. This concept is illustrated for a single dimension in figure 2, where the spring with stiffness k forms the suspension. The measurement stage is represented by the mass, m . Its acceleration, a_1 , is measured and used by a feedback controller, C , to produce an actuation force, F , that reduces the influence of the frame disturbance acceleration, a_0 , on the measurement stage.

The designed vibration isolation system is shown in figure 3. It consists of the measurement stage components (with all but the tube window omitted for clarity) mounted on a measurement frame. This measurement frame is mounted on a three degrees of freedom suspension (see figure 4), constraining three dimensions, while the other three dimensions are controlled by three voice coil actuators and three acceleration sensors. A degree of freedom (DOF) is an independent absolute or relative motion within the system describing relevant system motion. The three DOF not constraint by the suspension are the z-direction translation and rotations around the x- and y-axes. These are the first three eigenmodes of the suspension. All suspension frequencies are designed close to 30 Hz to keep the suspension design symmetric. The three actuators and sensors are used to close the feedback loops in the z-translation and x-axis rotation to provide active vibration isolation. The third DOF (the y-axis rotation) is used for the aforementioned measurement actuation.

With this setup it is also possible to investigate if it is sufficient to suspend and actively isolate only one direction, resulting in a more efficient solution. By attaching one or both optional constraints at the sides of the setup (see figure 3), the measurement frame can be constraint in its rotation about either or both the x- and y-axis in the suspension plane, reducing the system's number of DOF. The optimum configuration being a research objective. It might be possible

that one of the frame disturbances, a translation in y-direction, can cause significant unwanted motion in the tube window. Therefore an interchangeable four DOF suspension (see figure 4) is created to convert the setup into a three DOF active isolation setup with one passively suspended DOF.

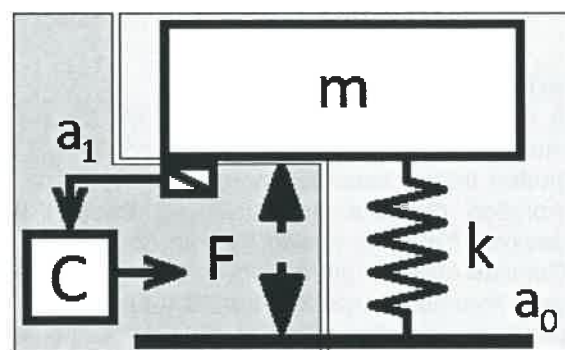


FIGURE 2. Dynamic model of suspended measurement stage (right block) with active feedback vibration isolation (left block).

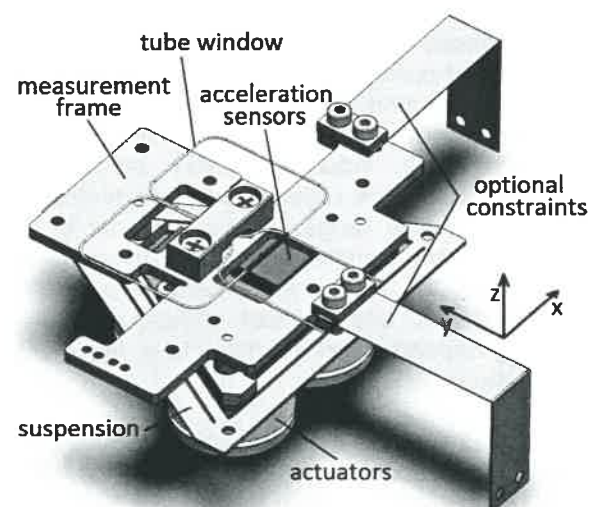


FIGURE 3. Solidworks® model of the active vibration isolation setup with optional constraints and simplified measurement stage.

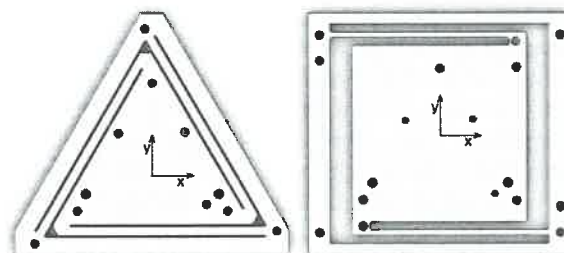


FIGURE 4. Solidworks® models of the 3 DOF suspension (left) and alternative 4 DOF suspension (right).

MODELING

The entire setup has been modelled using the non-linear finite element flexible multibody software package SPACAR [7]. This model (see figure 5) is used to obtain equations of motion for suspension design and transfer functions for control synthesis.

MODEL RESULTS

The three DOF have a suspension frequency between 30 and 33 Hz. The alternative four DOF suspension has an additional DOF in the y-direction, which is left passively suspended with a frequency of 34 Hz. The optional constraints are omitted in the rest of this work and will be used for future research.

Of primary interest is the transfer function from six (three orthogonal translations and three orthogonal rotations) frame acceleration disturbances to the tube window Coriolis-displacement at the sensing locations. This is called the transmissibility and it has six inputs and one output, thus it is a multiple-input single-output (MISO) system. As mentioned, the tube window displacement is mainly influenced by two measurement stage directions. Without taking the tube dynamics into account, the transmissibility of interest is thus the multiple-input multiple-output (MIMO) transfer function from the same six inputs to two stage acceleration measurements. These are the acceleration signal of the top sensor, $a_{1,I}$ and the mean of the bottom sensors, $a_{1,II}$, (see figure 5). These signals are a combination of the z-translation and x-axis rotation. From the new transmissibility only three frame accelerations appear to be significant due to symmetry, being a translational disturbance in z-direction, a translation in y-direction and a rotation around the x-axis. The other directions already have an influence below -100 dB. So only the three main transmissibility directions are considered when evaluating the performance of the setup.

For control synthesis the transfer function from actuator forces to measurement stage accelerations is required. This is called the plant and this transfer function has three inputs and three outputs. This transfer function has non-zero off-diagonal elements and thus coupling of every input to every output, where for control purpose it is desirable to have a diagonal matrix.

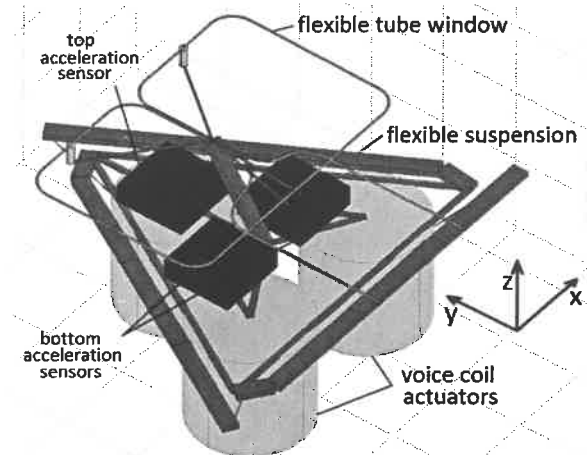


FIGURE 5. SPACAR model of the setup without optional constraints.

The plant transfer function can be decoupled by using the eigenvectors obtained from an eigendecomposition according to Owens [8]. This results in a predominantly diagonal transfer function matrix where influence of the off-diagonal elements is about 30 to 40 dB lower across the frequency region of interest. Three independent SISO plants remain for which three SISO controllers can be designed. The third decoupled plant direction is the transfer function from actuator input to sensor output that corresponds to the third DOF of the suspension, a rotation about the y-axis. In this direction no feedback loop will be closed as this DOF was only included to test tube window actuation. On the other two directions acceleration feedback will be applied. The applied SISO controllers consist of the aforementioned PI control together with three loop shaping filters. These add to the performance and stability of the feedback system. The additional filters used are a second order high-pass filter to limit actuator saturation, a second order low-pass filter to increase attenuation at the actuation frequency of 170 Hz and an zero at the crossover frequency of 300 Hz to increase stability. The parameters of the PI controller are calculated from performance requirements (lowering of the suspension frequency and adding damping) and the respective SISO plant transfer functions.

The three main transmissibility directions are shown in figure 6 for the reference, passive and active systems. The reference system is an unsuspended measurement stage, in which a y-translation has no influence on z-translation or x-axis rotation of the measurement stage. It can be seen that the attenuation at the measurement

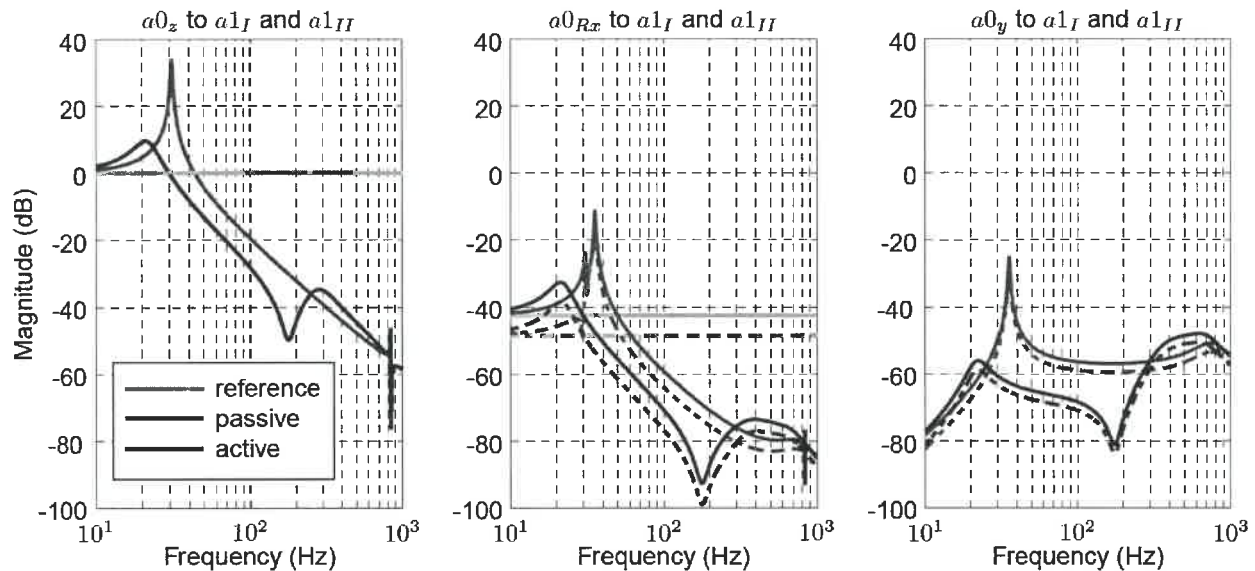


FIGURE 6. Bode magnitude plot of the main transmissibility directions (z-translation $a0_z$, -axis rotation $a0_{Rx}$ and y-translation, $a0_y$) to two measurement stage accelerations $a1_I$ (solid) and $a1_{II}$ (dashed).

frequency of 170 Hz for two of the three main directions is 50 dB with respect to the reference system. The third transmissibility shows that the vibration isolation system introduces the influence of a translational disturbance in y-direction at -80 dB.

EXPERIMENTAL RESULTS

The experimental setup consists of the suspended measurement stage, resembling the modelled setup of figure 3 when the optional constraints are omitted and the measurement stage components are added. This is mounted on a six DOF shaker platform used to supply external vibrations. An identification of the dynamics of this setup is performed over the frequency region between 5 and 2000 Hz by a modal analysis based on the method outlined by Tjepkema [9]. This results in three estimated decoupled SISO plants (shown in figure 7), similar to the plant obtained from the model and decoupling strategy. The main resonance frequencies in these transfer functions correspond to the three suspension frequencies, realized in the region between 25 and 34 Hz. An additional real pole is identified in each transfer function, originating from the (unmodelled) actuator induction. An internal mechanical mode is estimated at 282 Hz in one of the decoupled directions and in another directions an internal mode at 845 Hz is identified. Finally phase lag resulting from accelerometer signal filtering and digitization can be observed.

Based on the identified SISO plants the PI parameters are adjusted. The actuator induction poles are compensated by adding an extra zero and an additional notch filter is added at 845 Hz to stabilize the identified high frequency dynamics of the system.

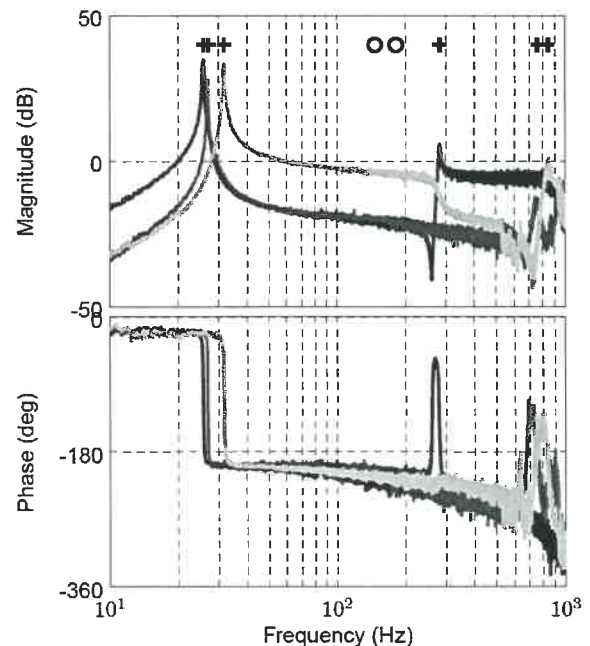


FIGURE 7. Identified plant (from actuator forces to three stage accelerations) with estimated complex pole pairs, o , and real poles, $+$.

As an initial validation only a disturbance in z-direction (the main disturbance direction as seen from figure 6) is introduced to the system. Both feedback loops are closed and performance is measured by evaluation of the power spectral density (PSD) plot of the measured stage acceleration signals (see figure 8). This PSD plot should resemble the modelled main transmissibility direction in the frequency region between 5 and 500 Hz as the input disturbance frequency content is uniform in this region.

The reference data is obtained by using the voice coil actuators to present a disturbance with an amplitude of $1 \times 10^{-4} \text{ (m/s}^2\text{)}^2/\text{Hz}$ to the measurement stage. In the PSD of this signal the suspension frequency is seen and the identified internal mode is present at the expected frequency of 285 Hz. The passive and active accelerations show the expected behaviour within the frequency region of interest and the attenuation of frame vibrations in z-direction for the passive and active systems is 35 and 48 dB in the relevant frequency region around the actuation frequency, respectively.

When the influence on mass flow measurement is evaluated, a time domain signal, as shown in figure 9 is obtained. No real mass flow is provided to the device, so the measurement is a flow error due to the vibration disturbance, scaled to arbitrary units. The attenuation of the

flow error obtained for the active system is lower than the expected 48 dB, due to an unexpected high measurement sensor noise. Larger disturbance amplitudes are provided to improve the signal to noise ratio, as seen from table 1, but the attenuation of 48 dB can still not be validated because the highest amplitude of $4 \times 10^4 \text{ (m/s}^2\text{)}^2/\text{Hz}$ is the upper limit of the used test setup.

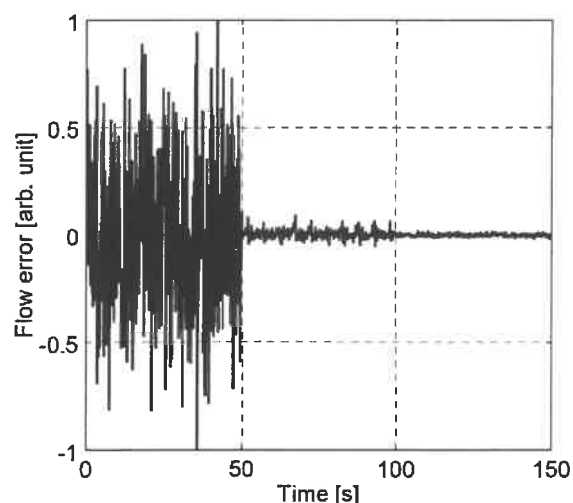


FIGURE 9. Time domain flow error with z-translation disturbance. Reference (0-50 s), passive (50-100 s), active (100-150 s).

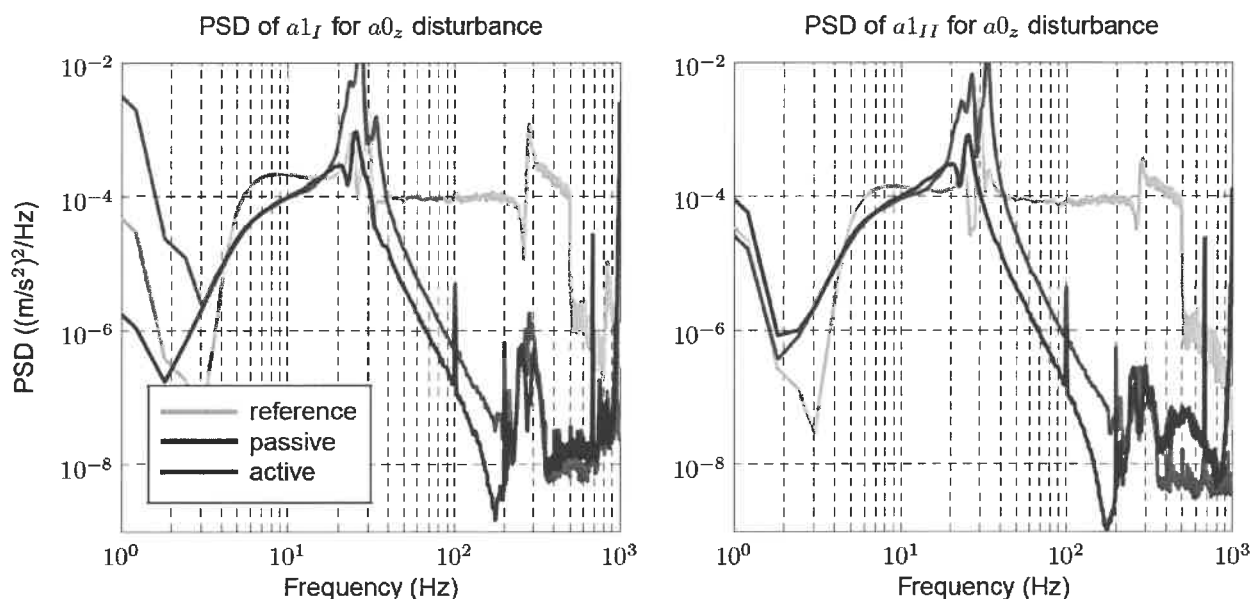


FIGURE 8. PSD of the two measurement stage accelerations for a translational disturbance (z-direction).

TABLE 1. Experimental RMS flow error in arbitrary units and the attenuation relative to the reference.

Disturbance (m/s ²) ² /Hz	Reference		Passive suspension		Active feedback	
	units	dB	units	dB	units	dB
1e-5	0.0551	0	0.0067	-18.53	0.0079	-17.44
1e-4	0.1741	0	0.0085	-26.28	0.0068	-28.44
4e-4	0.3515	0	0.0211	-24.41	0.0070	-34.03

DISCUSSION

To further identify the performance of this feedback system the influence of more disturbance directions (ultimately all six directions) will be tested. Also the optimal configuration of additionally constraint, passively suspended and actively isolated directions can be determined. The identification shows an internal mode at 282 Hz currently not seen in the dynamic model. Its origin should be identified, possibly modelled and its influence on control stability and performance should be further investigated. Furthermore robustification of the control for higher order dynamics of the setup should be investigated, for stability issues are encountered at higher specifications. The mentioned phase lag introduced by the digitization of the acceleration sensor signals has a negative influence on the system's stability and performance. It should be quantified and accounted for when determining obtainable specifications. Finally, in order to validate the attenuation of the influence of vibrations on the mass flow measurement a lower noise of the optical sensors used to measure tube displacement and thus mass flow is desired. The validation is currently limited to this noise level for low amplitude disturbances.

CONCLUSION

A valid conceptual design for the suspension and active vibration isolation of the Coriolis mass flow measurement stage in z-direction translation and x-axis rotation is created and three main disturbance directions are identified, being a translation in z- and y-direction and a rotation around the x-axis.

With a dynamic model of the presented setup design it has been shown that the theoretical attenuation of frame disturbances in the main directions is 50 dB.

Experimental validation of the active vibration isolation with feedback control shows an attenuation of about 48 dB when a disturbance is provided in the largest of the main disturbance directions, the z-direction translation.

ACKNOWLEDGEMENTS

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VIBRATION EFFECTS ON AN ENVIRONMENTALLY TOLERANT SCANNING WHITE LIGHT INTERFEROMETER

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ABSTRACT

Surface metrology instruments such as the scanning white light interferometer (SWLI) normally require thermal and vibration isolation and are therefore often located in appropriately controlled environments. A commercially-available "environmentally tolerant" SWLI suggests the possibility of shop-floor metrology, offering cost and process time savings. This paper describes a method for evaluating the vibration tolerance of such an instrument.

INTRODUCTION

To enable surface metrology in manufacturing environments, optical profilometry may not be the traditional choice due to inherent sensitivity to vibration [1]. Due to this limitation, such instruments are typically located in environmentally- and thermally-controlled environments. The effect of random vibration on optical measurements using phase-shifting interferometry is treated analytically in [2] and numerically in [3] for several common phase-shifting interferometry (PSI) algorithms. The 3, 4, 5, and 7-bucket PSI algorithms, all of which utilize $\pi/2$ spacing, exhibit similar frequency response to vibration along the optical axis; for a given disturbance amplitude, measurement error is a function of vibrational frequency, with local error maxima shown to be at odd multiples of half the instrument's detector frequency. Such error often appears as a ripple superimposed on the measured topography, occurring at twice the spatial frequency of the interference fringes. As the number of integrating buckets increases, sensitivity to vibration declines with increasing frequency. PSI algorithms using $\pi/3$ or other spacing also exhibit the twice fringe frequency ripple, but the frequency dependence differs from that of the $\pi/2$ algorithms.

Designed to enable metrology in uncontrolled environments, the Zygo ZeGage, shown in Fig. 1, is an environmentally-tolerant scanning white light interferometer (SWLI). SWLI algorithms have not been analyzed in the same detail as PSI

algorithms in the literature. They differ from PSI algorithms, but some laboratory instruments exhibit similar response to environmental disturbances. If the environmentally-tolerant instrument operates on the same principle as its laboratory counterparts, similar response characteristics can be expected. An analysis of this instrument's response to single-frequency vibration is presented, which follows a methodology applicable to any such instrument, regardless of manufacturer.

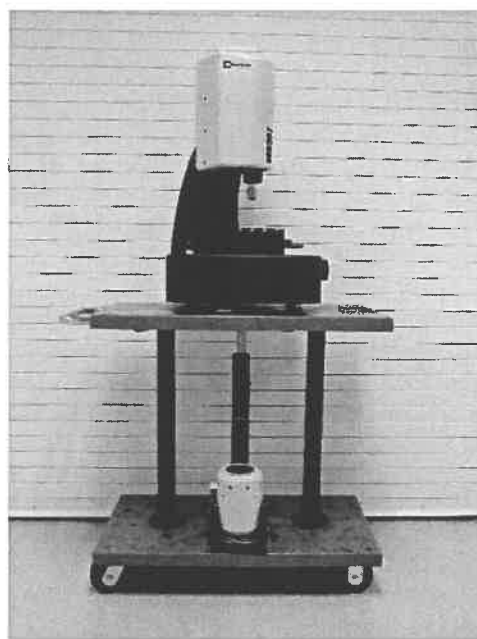


FIGURE 1. Zygo ZeGage SWLI

Measurements of sinusoidal and random-profile reference specimens using the ZeGage were shown in [4] to correlate well with stylus profilometer measurements. In [5], the instrument was placed on a laboratory cart and excited using an electrohydraulic shaker mounted rigidly to the cart base. When excited at frequencies near a structural resonance, RMS measurement error (the difference between measurements during

static and excited conditions) was demonstrated to increase due to relative motion between the specimen and the instrument objective. Figure 2, from [5], shows the frequency dependence of specimen-objective motion when the instrument as shown in Fig. 1 is excited with the base-mounted shaker.

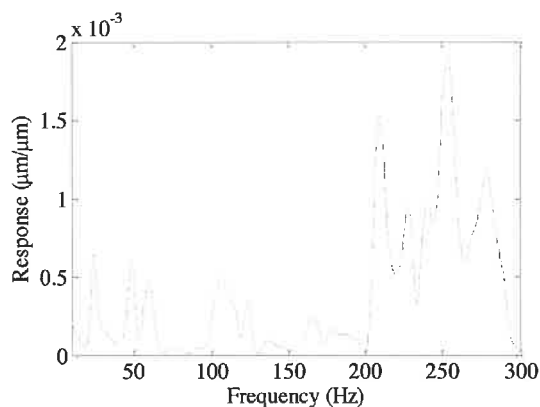


FIGURE 2. Relative specimen-objective motion in μm per μm of input vibration at cart base.

EXPERIMENTAL TECHNIQUE

From the results shown in [5], a correlation between frequency and amplitude of relative specimen-objective motion was suggested. Such an analysis should be similar in form to the experimental validation in [2]. The structural dynamics of the instrument prevent the generation of single frequency uniaxial vibration using a remotely-mounted shaker. Aside from this limitation, such shakers are generally inappropriate for generating vibration amplitudes at or below the order of one fringe. To overcome this limitation, flexure-based specimen shakers are described that oscillate the sample directly beneath the objective along (shown in Fig. 3) and perpendicular to the instrument's optical axis (shown in Fig. 4). These specimen shakers are actuated by water-cooled preloaded PZT stacks and are designed to facilitate rigid attachment to the instrument's specimen stage. When driven with a function generator through a 20 dB attenuator, motion amplitudes from 10 nm to 300 nm are achievable over a wide frequency range.

A polished mirror specimen was oscillated along (vertically) and normal to the optical axis (horizontally) at amplitudes from 12 nm to 125 nm, corresponding to approximately one-half fringe at a nominal 500 nm wavelength, at frequencies from 10 Hz to 300 Hz. At each

increment of 2 Hz and 6 nm, the sample was measured six times using a 50x objective.

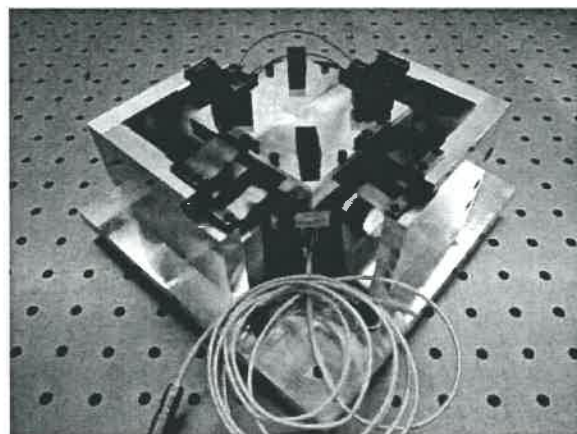


FIGURE 3. Specimen shaker for oscillation along instrument optical axis.

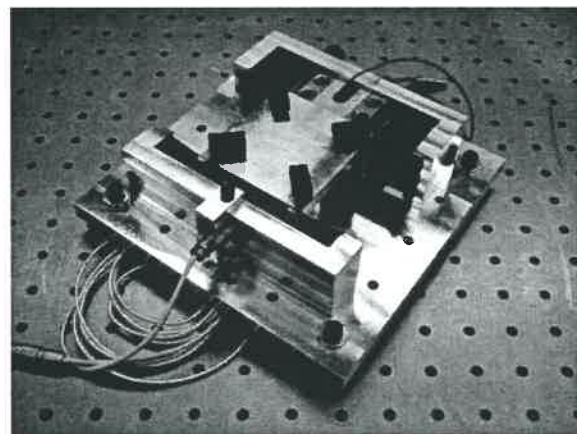


FIGURE 4. Specimen shaker for oscillation normal to instrument optical axis.

Figures 5-6 show the interference fringe orientation for the static condition with no forced vibration and the measured topography under 125 nm sinusoidal vibration at 250 Hz. The twice fringe frequency ripple discussed by de Groot [2] is clearly evident; 3 interference fringes are visible in Fig 5, corresponding to 6 peaks in the ripple pattern seen in Fig 6.

Power spectral density (PSD) analysis was performed for each amplitude and frequency combination; the difference in quadrature between the bandwidth-limited spatial frequency content of the measured surface and a reference measurement at static condition yields an RMS measurement error. The results of this test are shown as a surface plot in Fig 7. A similar set of experiments was performed using the polished

mirror specimen and the horizontal specimen shaker; results are shown in Fig. 9.

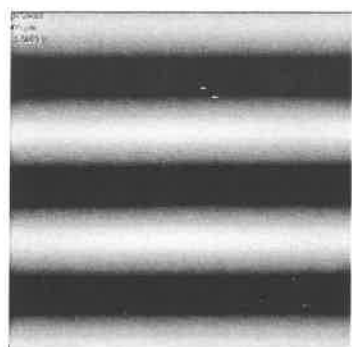


FIGURE 5. Interference fringe orientation at static condition (no forced vibration).

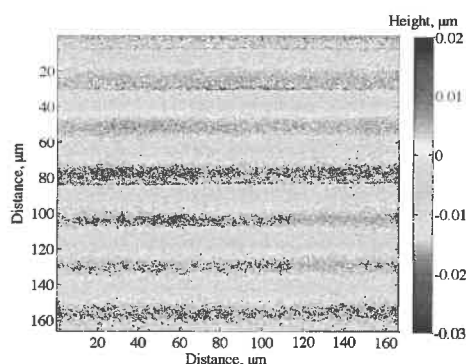


FIGURE 6. Measured surface topography under 125 nm, 250 Hz sinusoidal vibration.

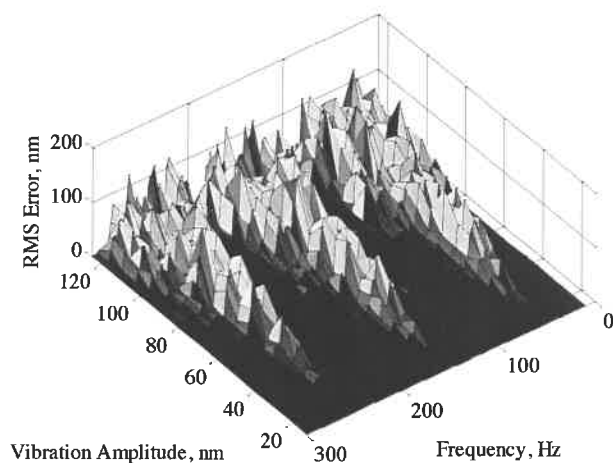


FIGURE 7. RMS error of polished mirror specimen oscillated along instrument optical axis.

Examination of Fig. 7 suggests a general increase in RMS error with increasing vibration amplitude; as predicted in [2], local error maxima occur at odd half-multiples of the detector frequency, 100 Hz for this instrument (maxima occur at 50 Hz, 150 Hz, 250 Hz). As vibration amplitude increases, the spread of the error about these maxima broadens substantially.

This behavior is consistent with that of an instrument using the three-bucket PSI algorithm described in [2]. Under horizontal vibration (Fig. 9), error magnitudes (attributable to image blurring) are negligible compared to the vertical case, with no significant dependence upon vibrational frequency or amplitude.

The experiments were repeated using an electroformed nickel sinusoidal reference specimen (Rubert 528, 50μm wavelength, 1μm peak-valley height). Similar results to those with the polished mirror specimen are shown in Figures 8 and 10. Similar error magnitudes occur for the case of vertical oscillation, suggesting an instrument response independent of surface measured. Higher error magnitudes are evident for the case of horizontal oscillation; they are attributed to the presence of this specimen's more complex surface features.

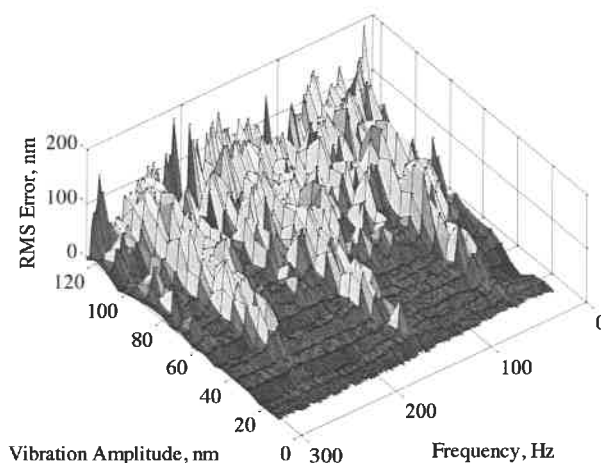


FIGURE 8. RMS error of sinusoidal reference specimen oscillated along instrument optical axis.

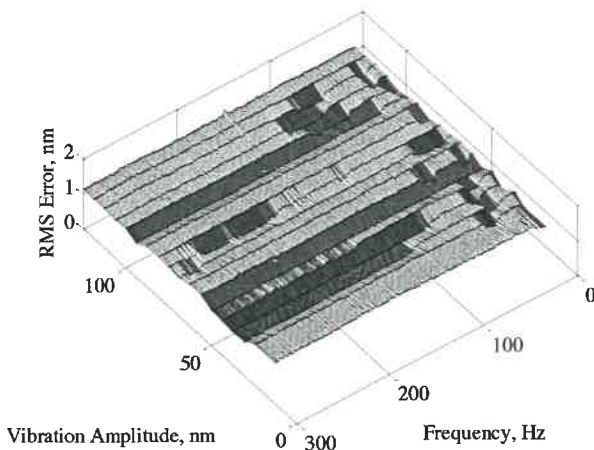


FIGURE 9. RMS error of polished mirror specimen oscillated normal to optical axis.

REMARKS

For single-frequency vibration of a specimen along the instrument's optical axis, local error maxima exist at odd half-multiples of the instrument's camera frequency, as predicted by de Groot in [2]. Error magnitudes, as well as the width of the error envelope, increase with vibration amplitude. For vibration normal to the optical axis, errors are primarily due to blurring; no dependence upon frequency or amplitude is suggested. Specimens with more complex surface form exhibit higher errors than those nearly void of form.

Operating environments for metrology instruments are often specified in reference to standardized Vibration Criterion (VC) environments, as described in [6]. Zygo recommends that the ZeGage be installed in an environment conforming to VC-C or better, corresponding to a maximum vibration velocity of $12.5 \mu\text{m/s}$ for all frequencies between 8 Hz and 80 Hz. Following Fig. 2, for the instrument and cart setup in Fig 1, such an environment would cause maximum specimen-objective motion of 0.04 nm amplitude at 25 Hz. As specimens with more complex surface form are found more prone to measurement error, following Fig 9, for RMS measurement errors to remain negligible, specimen-objective motion amplitude should be kept well below the lowest tested level. A somewhat arbitrary choice of 5 nm maximum specimen-objective motion would correspond to a permissible broadband level of $1275 \mu\text{m/s}$, well above the manufacturer-specified VC-C level as

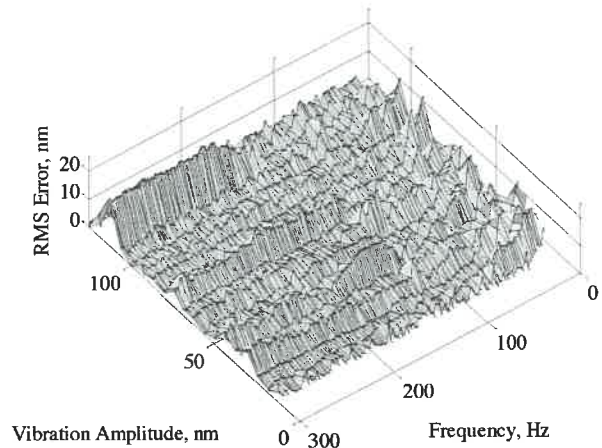


FIGURE 10. RMS error of sinusoidal reference specimen oscillated along instrument optical axis.

well as the ISO-specified 'workshop' level indicated in [6]. Environments described by this 'workshop' level or better are likely suitable for desirable operation of the ZeGage instrument.

Although this evaluation was performed on a specific instrument, the technique is applicable to any surface metrology instrument. Further investigation of instrument response to broadband disturbances more typical to an industrial environment would also be beneficial in specifying such an operating environment.

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REPRODUCIBILITY AND DYNAMIC STABILITY OF AN ABBE-COMPLIANT LINEAR ENCODER-BASED MEASUREMENT SYSTEM FOR MACHINE TOOLS

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INTRODUCTION

Abbe-compliant machine tool concept with linear encoders

The machining accuracy in machine tools is mostly determined by the positioning accuracy of the tool with respect to the workpiece. The largest positioning errors are due to two error sources: (1) thermo-mechanical errors, such as thermal deformation and expansion of the structural frames, and (2) geometric errors, which arise predominantly by Abbe errors caused by angular error motions of the slides together with offsets between the position measurements and the point of interest, in this case the tool. In state-of-the-art machine tools, thermo-mechanical errors are reduced by extremely stable temperature control of the surroundings and/or thermal modeling and compensation. Lowering of the geometrical errors on the other hand is achieved by the application of air bearings or hydrostatic bearings and volumetric calibration of the machine tool axes.

At KU Leuven, a new concept to further increase machining accuracy in machine tools has been proposed. In this concept, the structural loop and the metrology loop are functionally separated [1] and the position of the workpiece is measured with linear encoders according to the Abbe principle in multiple degrees-of-freedom [2]. Figure 1 shows the conceptual layout of such a machine tool with the so-called 'moving-scales'. A tool metrology frame (TMF) is kinematically connected to the tool spindle, while workpiece metrology frames (WMF) surround the workpiece. A master metrology frame (MMF) holds short stroke sensors that monitor the movement of the TMF due to thermo-mechanical errors or deflection of the structural frame. It also contains the reading heads of the linear encoders that track the movement of the WMF. This configuration ensures that no forces enter the metrology loop and allows for

independent optimization of the structural frames and the metrology frames. Tracking of the WMF movement by the moving-scales is performed by locating a short stroke sensor, such as a capacitive sensor, at the tip of and in-line with the scale. The short stroke sensor detects the movement of the WMF target surface and a linear motor drives the carrier holding the scale and the sensor to keep the short stroke sensor within its measurement range. The WMF position measurement then consists of the addition of the short stroke sensor measurement and the linear scale measurement. This configuration hence allows the scales to always be pointed to the tool center and, theoretically, the Abbe errors can be reduced to zero, which drastically improves the motion accuracy of the machine tool.

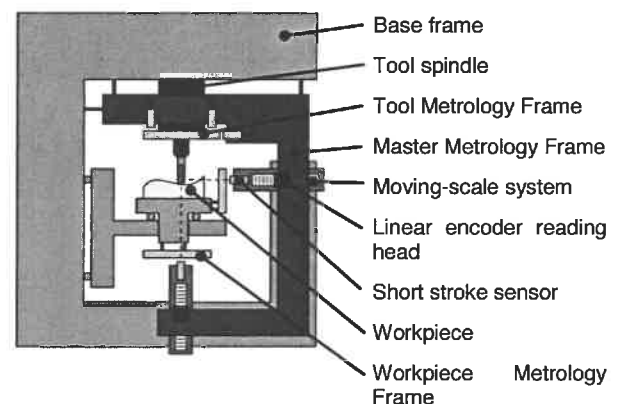


FIGURE 1. Machine tool concept with functional separation of the structural and the metrology loop and Abbe-compliant linear encoder configuration.

1-DOF moving-scale measurement system

A 1-DOF prototype of a moving-scale system with a measurement length of 120 mm is depicted in Figure 2. The capacitive sensor is mounted on a stainless steel interface with the same coefficient of thermal expansion as the sensor, which is in turn kinematically connected

to the scale carrier. The Zerodur® scale (Heidenhain LIP 281) is kinematically constraint to the scale carrier and the sensor interface by using the thermal centre principle in such a way that expansion of the sensor or the scale carrier causes no displacement of the scale w.r.t. the sensor tip [3]. The scale carrier is connected to a moving base frame that holds a linear recirculating ball bearing and a linear motor. The reading head of the linear encoder, which is not shown in this figure, is connected to the metrology frame.

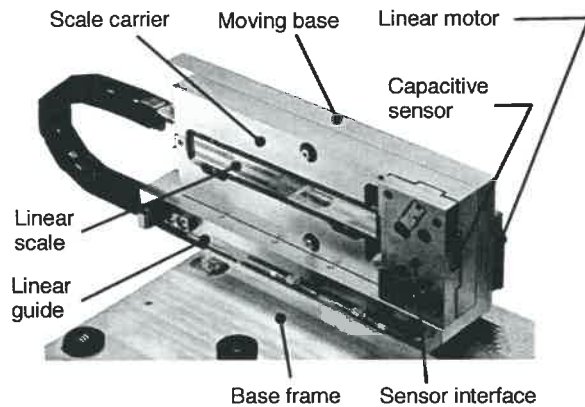


FIGURE 2. Layout of the 1-DOF moving-scale prototype. The reading head is not shown.

All components contributing to the measurement uncertainty have been determined and are summarized in Table 1. The uncertainty budget can be split up into three parts. The first part consists of the error components that are

caused by changes of the environmental conditions, such as temperature, relative humidity and air pressure, and amounts to 14 nm for $\pm 0.5^\circ\text{C}$ temperature changes and $\pm 5\%$ humidity changes. The second part involves more rapidly changing errors such as non-repeatable angular error motions of the guides that cause Abbe-errors and dynamic errors due to movement of the components and sensor noise. The last part of the budget contains the length uncertainty of the scale after calibration and the calibration uncertainty of the capacitive sensor. The scale will be calibrated by a laser interferometer; then, the capacitive sensor is calibrated on the machine using the linear scale measurements to also account for non-linearity effects induced by non-perpendicularity of the sensor w.r.t. the target surface. The total measurement uncertainty for the 120-mm moving-scale system is 21 nm ($k=2$).

REPRODUCIBILITY OF MOVING-SCALE MEASUREMENT SYSTEM

System layout and uncertainty budget

In order to verify the reproducibility and the repeatability of the moving scale system under changing environmental conditions, the measured displacements should be compared to the measurements of a stable reference that is placed in line with the moving scale system. For example, the reference could be a moving

TABLE 1. Uncertainty budget for a 120 mm 1-DOF moving-scale measurement system.

Component	$U_{(k=2)}$ [nm]
Affecting reproducibility (for changing environment $\Delta T = \pm 0.5^\circ\text{C}$ and $\Delta RH = \pm 5\%$)	14
Scale expansion (CTE Zerodur® = ± 0.1 ppm)	12
Reading head thermal drift	3
Scale carrier assembly thermal drift	4
Capacitive sensor temperature and humidity drift after compensation	2
Other errors due to changes in environmental conditions	5
Repeatability	8
Non-repeatable Abbe errors	1
Dynamic errors	5
Sensor noise	5
Calibration and alignment uncertainty	13
Abbe errors	4
Cosine error	2
Linear scale calibration	12
Capacitive sensor non-linearity calibration	2
Combined measurement uncertainty	21

mirror, where the moving-scale is placed on one side of the mirror and a laser interferometer on the other side of the mirror. However, the laser beam path should then reside in a separate stable enclosure in order to reduce the influence of the changing environment in which the moving-scale system is operating. It is very difficult and expensive to create an enclosure for an interferometer which is stable within less than 20 nm over a measurement length of 120 mm while the surrounding environment exhibits relatively large ($\pm 0.5^\circ\text{C}$) temperature variations. Nevertheless, to determine reproducibility, it is not important to trace the measurements back to the length standard. Therefore, the use of a second stable reference scale and elimination of any systematic errors due to grating errors of the scale and repeatable Abbe and cosine errors from the measurements are allowed.

For this reason, a comparison of the moving-scale measurements to the measurements of a similar system, called 'reference-scale', was opted for. The reference-scale is a mirrored copy of the moving-scale system, except that the capacitive sensor and its interface are replaced by a target surface. The system is shown in Figure 3 and has been discussed in [3]. The reading heads of the encoders are kinematically connected to an aluminum metrology frame that is in turn constrained to the base frame in a kinematic way using ball-in-V grooves. The reference-scale's position is monitored by two reading heads. Because the expansion of the Zerodur® scale is assumed to be invariant to temperature changes, the difference in readout of the two reading heads equals the expansion of the metrology frame over the distance L_2 (55 mm). If one assumes a uniform expansion of the metrology frame, which is a reasonable assumption for aluminum because of its high thermal diffusivity, the expansion between the reading heads of the moving-scale and the reference-scale over the length L_1 (230 mm) can easily be calculated.

The measurement error, of which the variance determines the reproducibility and the repeatability of the setup, is defined as

$$e = M_{MS} - M_{Rep} \quad (1)$$

with

$$M_{MS} = M_{RH1} - M_C \quad (2)$$

$$M_{Rep} = M_{RH2} + \frac{L_1}{L_2}(M_{RH2} - M_{RH3}) \quad (3)$$

and in which M_{RH1} , M_{RH2} and M_{RH3} are the read-outs of the reading head of the moving-scale and the two reading heads of the reference-scale respectively, M_C is the read-out of the capacitive sensor and L_1 and L_2 are the distances between the reading heads as indicated in Figure 3. M_{RH1-3} are positive when the system is moving in the positive x-direction. M_C is defined positive if the moving-scale and reference-scale are approaching one another.

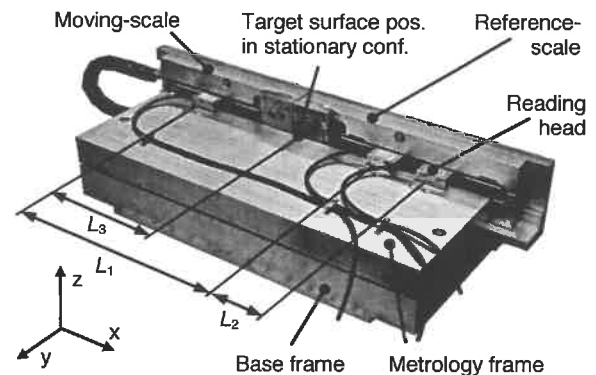


FIGURE 3. Setup for determining the reproducibility of the 1-DOF moving-scale measurement system in the movable configuration.

To eliminate the measurement uncertainty of the reference scale and to reduce the uncertainty associated with the metrology frame expansion compensation, the reproducibility setup can also be configured in such a way that the moving-scale measures a stationary target attached to the metrology frame. The two reading heads of the reference scale are still used for compensation of the thermal expansion, but the length of the metrology frame that requires compensation is now the shorter distance between the moving-scale reading head and the stationary target surface, which is L_3 (123.5 mm). Eq. (3) then reduces to

$$M_{Rep} = \frac{L_3}{L_2}(M_{RH2} - M_{RH3}) \quad (4)$$

The uncertainty budget of the reproducibility of the setup in both configurations is shown in Table 2. While in the movable configuration, the uncertainty is dominated by the uncertainty of the expansion L_1 compensation, this value is significantly reduced in the stationary target

TABLE 2. Uncertainty budget of the setup for determining the reproducibility of the 1-DOF moving-scale for temperature changes of $\pm 0.5^\circ\text{C}$ and relative humidity changes of $\pm 5\%$.

Component		$U_{\text{movable}} (k=2) [\text{nm}]$	$U_{\text{stationary}} (k=2) [\text{nm}]$
M_{MS}	Moving-scale reproducibility U_{MS}	14	14
	Reference-scale reproducibility: U_{RS}	19	0
	Reading head stability: U_{RH}	3	3
	Scale expansion over L_2 : U_{SC}	6	6
M_{Rep}	Expansion L_2 : $U_{L_2} = \sqrt{2 \cdot U_{RH}^2 + U_{SC}^2}$	7	7
	Expansion $L_{1/3}$: $U_{L_{1/3}} = L_{1/3}/L_2 \cdot U_{L_2}$	29 (L_1)	15 (L_3)
	Uncertainty of reference position U_{Rep}	34	15
Combined uncertainty: $U_{tot} = \sqrt{U_{MS}^2 + U_{Rep}^2}$		37	21

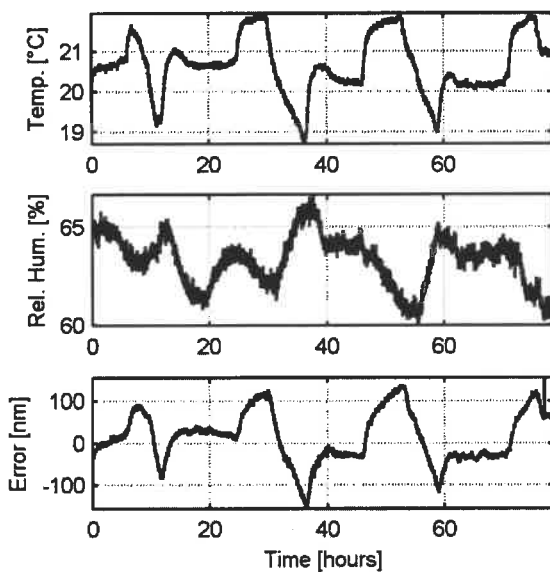


FIGURE 4. Temperature, relative humidity and measurement error (Eq. (1)) for the setup in the movable configuration.

configuration. However, the movable configuration will still be used to determine the expansion of the scale due to its mounting and to determine the repeatability of the system during movement.

Experimental verification

The setup of Figure 3 has been placed in a temperature controlled environment of $20 \pm 0.5^\circ\text{C}$. Air flow from the vents of the room's air conditioning system could cause a thermal gradient in the metrology frame. Therefore, an additional aluminum shielding with a thickness of 5 mm has been placed over the metrology frame. Next, a PVC enclosure of dimensions 650 mm $\times$ 650 mm $\times$ 500 mm has been placed over the entire setup to filter fast temperature

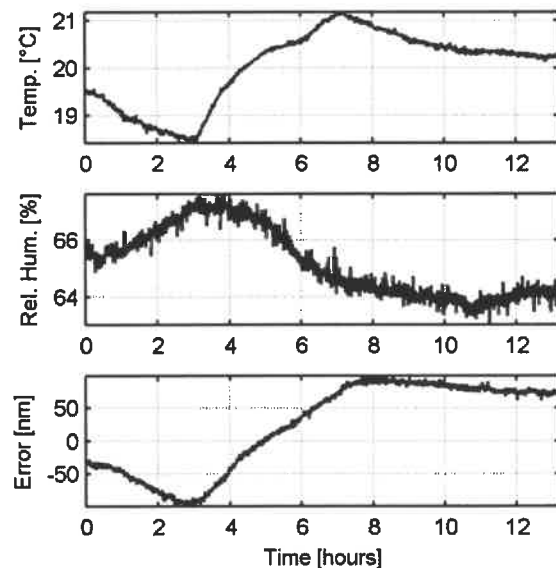


FIGURE 5. Temperature, relative humidity and measurement error (Eq. (1)) for the setup in the stationary configuration.

fluctuations in the room. Temperature and humidity have been monitored and the measurement error has been determined using Eq. (1) to (4).

Figure 4 shows measurement graphs for the setup in the movable configuration. The slides have been locked in position in this experiment in order to eliminate additional dynamic errors, which will be described in the next section. Figure 5 gives the same data for the stationary configuration with a stationary target attached to the metrology frame. The temperature and humidity dependence have been determined by a linear least squares estimate on the data from which slow non-linear drift and high-order effects were filtered out (Table 3). The table indicates

that for the temperature sensitivity, measurements are fairly corresponding to what has been predicted in Table 2. Further experiments should identify the temperature sensitivity of each component in the setup. On the other hand, the humidity sensitivity is larger than expected. The only component that up till now has been assumed to drift with humidity changes is the capacitive sensor, but this humidity drift has been compensated by using a second capacitive sensor as a weather station [4]. It has been shown that by doing so, the humidity drift could be almost completely eliminated. Further research should identify the source of the humidity drift and attempt to reduce it in order to attain the envisioned measurement uncertainty.

TABLE 3. Temperature and humidity sensitivity of the reproducibility setup. Between brackets the resulting expanded uncertainty is indicated.

	Movable configuration	Stationary configuration
Temperature sensitivity [nm/°C]	62 ($U_{(k=2)} = 36$)	47 ($U_{(k=2)} = 27$)
Humidity sensitivity [nm/%]	-18 ($U_{(k=2)} = 10$)	-19 ($U_{(k=2)} = 11$)

DYNAMIC STABILITY

As explained before, the moving-scale system tracks the movement of the target surface by controlling the linear motor in such a way that the gap measured by the capacitive sensor stays within its measurement range. To design an optimal controller, first the transfer function of the moving-scale from linear motor current to capacitive sensor output has been determined. Due to the range limit of $50\text{ }\mu\text{m}$ of the capacitive sensor, the linear scale was used as the output in the identification process. The frequency response function (FRF) and the least-squares estimate (LSE) of this transfer function are shown in Figure 6. The first resonance frequency varies from 189 Hz to 242 Hz, depending on the position of the moving scale, which is due to the changing moment of inertia. This low eigenfrequency and the pole-zero characteristic of the system that makes the phase drop below 180° after the first resonance, limits the bandwidth of the control loop. This control loop was designed with a PI velocity control on the output of the linear encoder and a PI2 position control on the output of the

capacitive sensor. Additionally, a notch filter has been inserted around the first eigenfrequency. The bandwidth of the control loop is 60 Hz. The largest disturbance force the system has to overcome is the friction in the guides ($<6\text{ N}$). Nevertheless, Figure 7 shows that this simple control strategy is feasible to keep the capacitive sensor gap within $\pm 25\text{ }\mu\text{m}$ while tracking the reference-scale.

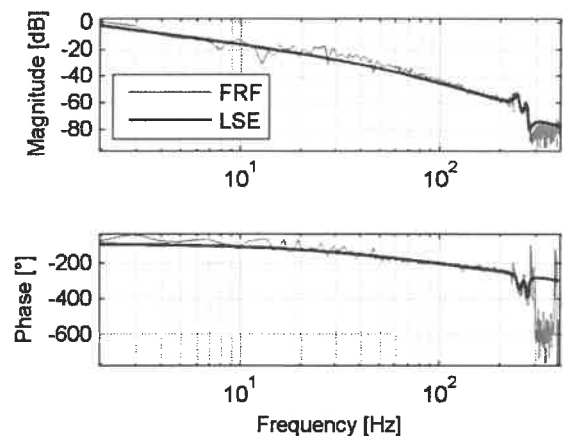


FIGURE 6. Frequency response function (FRF) and least-squares estimate (LSE) of the transfer function from motor current to moving-scale position.

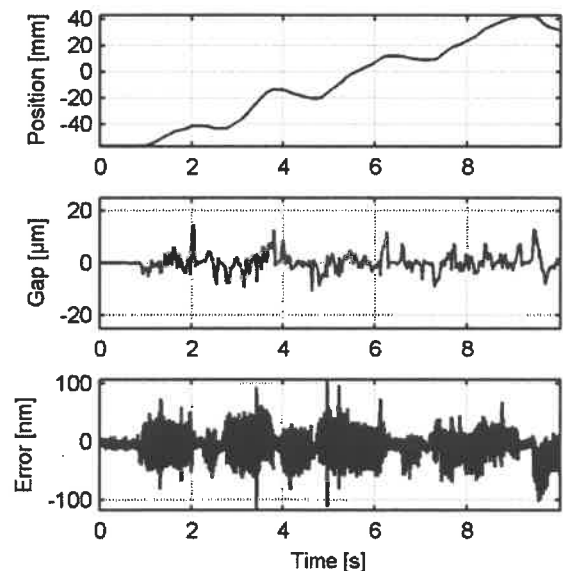


FIGURE 7. Position of moving-scale, capacitive sensor gap measurement and measurement error (Eq. (1)).

Figure 7 also shows that while the system is moving, the measurement error amounts to 50 nm. This error consists of high frequency components and static reversal errors. The latter

is due to the limited stiffness of the guides and the fact that the setup has not been aligned to reduce the Abbe offset and the non-perpendicularity of the capacitive sensor w.r.t. the target surface. [5] proposes a method for alignment of the moving-scale system.

For the high-frequency components, the errors were found to be caused by high-frequency accelerations caused by a pitching and yawing motion of the guides due to stick-slip. The kinematic connection between the capacitive sensor holder and the scale has a limited stiffness and although the first eigenfrequencies of the scale carrier assembly are above 1600 Hz, a finite element analysis revealed that an acceleration of 1 m/s^2 causes the distance between the scale and the probe to reduce with 20 nm by compliance of the Hertzian contacts. This value has been experimentally verified by making the moving-scale oscillate at 40 Hz and calculate the sensitivity of the measurement error to the moving-scale accelerations (Figure 8). The calculated value of 19 nm corresponds very well with the one resulting from the model. Next, experiments with the moving-scale and reference-scale connected to each other and both moving at constant speed have been performed, from which the same value for the acceleration dependence could be deduced. It can therefore be concluded that the relative displacement of the capacitive sensor w.r.t. the scale due to high-frequency accelerations results in the high-frequency measurement error. To reduce these errors, the recirculating ball bearings of the current system will be replaced with roller bearings that have very low friction and a much higher angular stiffness.

CONCLUSION

The performance of an Abbe-compliant linear encoder-based measurement system for machine tools called 'moving-scale' has been presented. A dedicated setup to determine the reproducibility and the repeatability has been built and experiments have been conducted. The measured temperature sensitivity showed good agreement with the predicted values, but the source of significant humidity drift should still be identified. Initial experiments on the dynamic performance made clear that stick-slip behavior in the guides, together with limited stiffness of the kinematic connections between the sensor and scale gave rise to large measurement

errors. In the future, these errors can be reduced by installing low friction linear roller bearings.

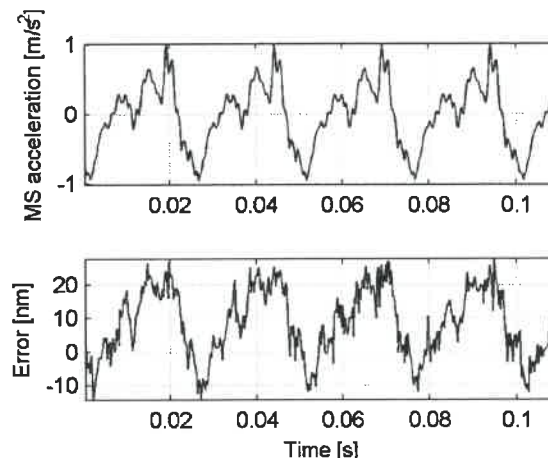


FIGURE 8. Acceleration of the moving-scale by a 40-Hz oscillation with $9 \mu\text{m}$ amplitude and corresponding measurement error (Eq. (1)). The measurement error w.r.t. acceleration is estimated to be $19 \text{ nm}/(\text{m/s}^2)$.

ACKNOWLEDGEMENTS

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THERMAL ERROR COMPENSATION FOR LARGE HEAVY DUTY MILLING-BORING MACHINES

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INTRODUCTION

Large heavy duty milling and boring machines are commonly used in demanding sectors such as oil&gas, wind energy, aeronautics, etc. Heavy duty operations lead to high heat dissipation within the machine, and this combines with the ever increasing accuracy requirements in these high added value applications. Therefore, machine builders need to implement smart thermal management strategies to be able to meet customer demands. The general challenge is described here, and a thermal error compensation system developed for large milling and boring machines is presented next. Finally, experimental results are shown to demonstrate the effectiveness of the system.

THERMAL ERRORS IN MACHINE TOOLS

Temperature variations within the machine cause thermoelastic deformations resulting in geometric inaccuracies in the workpiece, contributing more than 50% of the overall error [1]. Temperature variations are induced by a number of heat sources, both internal and external to the machine: ambient temperature and sun radiation; heat dissipated in the motors, bearings and gears; hot chip and heat in tool and workpiece produced in the cutting process; cooling fluids within structural elements or in the cutting area, etc.

The variety of relevant heat sources require thermal effects be considered at every step of machine tool lifetime, from design to daily operation. SORALUCE's milling and boring machines (see Figure 1) are designed for heavy duty production of large and complex parts, which typically require a large workspace, very high cutting forces, use of multiple tools, spindle heads and quills, long processing times and large machine workspace. Vertical travels up to 8m and horizontal in tens of meters are common requirements. Heat generation is high considering that spindles can rate up to 88 kW power, and thermal errors are magnified by the large dimensions of the machine.

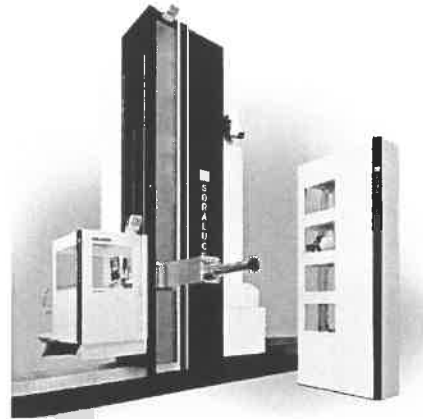


FIGURE 1. SORALUCE milling-boring machine with automatic spindle/quill changer

In these machines, the heat generated by the spindle rotation, in bearings, gears and motor itself, is the largest contributor to thermal errors, but other effects such as ambient temperature, motors in linear axes, workpiece temperature or the hot chip possibly falling against the machine can be relevant in some circumstances and may require specific developments.

Thermal Error Management

Three main strategies can be distinguished for thermal error reduction:

Machine Temperature Control

Limiting temperature variations is the most straightforward way to reduce thermal errors. Cooling circuits embedded in critical machine locations are used to remove heat from the machine to reduce temperature variations that induce errors.

Temperature control is necessary and effective, but it faces strong limitations due to the high generated heat in machines like the ones studied here, limitations to access critical points (e.g. rotating elements), etc, and thus relevant temperature variations must be expected in

heavy duty operations. The goal of the cooling system is thus not to keep the machine at 20°C (which is not possible), but to keep the temperature within a reasonable range and with slow variations.

Design for Thermal Error Reduction

A second approach is to minimize the deformations generated by the temperature variations. Machine design concepts such as symmetry can reduce the thermal errors induced by temperature gradients within the machine.

However, thermal design principles cannot always be applied to their full extent, being in conflict with others leading to high stiffness for high productivity or cost reduction, for example. A reasonable trade-off is to focus on optimizing the design to avoid angular errors, which are more difficult to compensate.

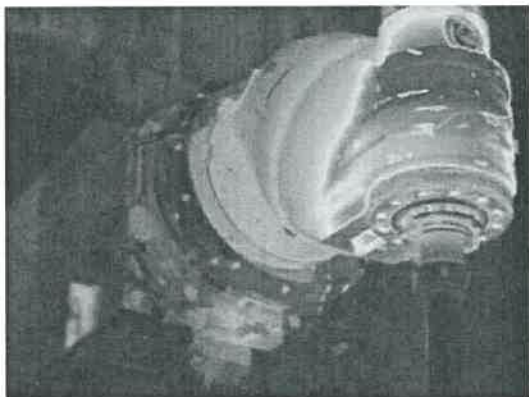


FIGURE 2. Thermal image of spindle head

Error Compensation

The third strategy is to develop thermoelastic models that can be calibrated to predict errors during machine operation and compensate them in the machine position control. Once developed, it is a relatively simple and low cost solution, since it only involves some machine occupation time and changes in the CNC software.

Given the limitations of the previous approaches, thermal error compensation is a meaningful solution to improve machine accuracy. Some background on this technology is provided next.

Thermal Error Compensation

Thermal error compensation has been a topic of research for already many years [1]. The main idea is to implement a model that predicts in real time the instantaneous error of the machine, and to compensate the position of the axes of the machine accordingly. A number of strategies have been proposed, with different approaches to machine characterization, modeling and implementation [2].

Machine characterization

The main goal here is to acquire information on the thermoelastic behavior of the machine in a wide range of working conditions, and to use this information for feeding the thermal error model that is used for predicting the model.

The characterization is typically performed by means of experimental tests on the machine, analyzing the influence of the ambient temperature, spindle speed rotation and linear axis motion, etc. The thermal error is measured either with displacement sensors measuring against a tool or other calibration artifacts, or by machining a part and later measuring it on a CMM. Minimization of the machine occupation time is a key aspect to reduce the costs, considering that thermal effects are usually slow.

Other authors propose methods based on virtual models of the machine to avoid the need to test the machine [3]. This approach seems however limited, since heat sources such as friction on bearings, or natural convection in the machine structure seem difficult to model accurately. Virtual thermoelastic models are useful at design stage for minimizing machine deformations, and for selecting the optimal location for temperature measurement. However, testing each machine individually adds limited extra cost and ensures highest accuracy.

Thermal error models

The information gathered during the machine characterization needs to be compiled in a simulation model which can be implemented in the CNC of the machine. Several models have been proposed in the last years to estimate in real time the thermal error of the machine.

A first classification can be made in function of the inputs to the model. Some authors have developed models using process information available at the CNC, such as spindle and axis speed, power, etc [4]. In some cases ambient

temperature data is also used. The alternative is to use of information of temperature sensors embedded in the machine.

The second classification can be made according the mathematical approach used in the model. Multiple linear regression models are used to obtain linear static parametric models that can easily be implemented in the machine control [5–7]. Dynamic models, in form of transfer functions, have been proposed too, in order to account for the observed delays between input variables and errors [8]. More complex mathematical solutions have been proposed too, with Support Vector Machines [9], Artificial Neural Networks [5,10,11], Adaptive Neuro Fuzzy Inference Systems [11], etc.

Implementation

Thermal errors vary continuously with machine operation, and therefore the compensation needs to be calculated in real time in the machine. In past years, with closed control architectures, options were mainly limited to either basic PLC programming, which permitted only simple models, or adding industrial PCs only to calculate the compensation.

In the last years CNC manufacturers are implementing more open architectures that allow more complex functions to be implemented in the control by third parties. This now allows the industrial implementation of most models that can be developed at research level.

NEW THERMAL COMPENSATION SYSTEM

Thermal error compensation offers a comparatively low cost solution for significantly improving machine accuracy, but the development of a reliable model and the practical implementation pose many challenges. A semi-automatic thermal characterization and compensation solution has been developed for SORALUCE milling and boring machines, with accuracy, but also industrial feasibility in mind.

Requirements

The goal of this project was the development of a thermal compensation system that could be applied to the full range SORALUCE milling-boring machines without requiring advanced operator skills. These are the main characteristics of the system:

- Characterization of thermal errors at Tool Center Point (TCP) in XYZ directions,

considering their variation with RAM and quill length and head orientation with few temperature sensors embedded in the machine.

- Fast machine characterization and with a simple setup, minimizing machine occupation time, considering that such machines typically have multiple spindles and quills, which need to be compensated separately.
- Automatic model identification from measured data and implementation compatible with four different CNCs (Heidenhain, Siemens, Fanuc and Fagor).
- The compensation should improve the accuracy of the machine, but almost most importantly, it should be robust to improve accuracy in all machine operating conditions.

Thermal Error model

The several alternatives proposed in the literature have been discussed above. For this project, the choice has been based on the following concepts:

Model inputs

Temperatures measured in key structural elements of the machine are considered here as only inputs to the model, since the temperature field is directly correlated to the thermal growth. Other inputs, such as process data and ambient temperature, are considered as heat sources, which can be used for predicting the temperature field with more complex models and higher uncertainty. The approach is thus to use whenever possible only direct temperature readings as input to the model.

Model structure

In an ideal case, a perfect model could be built knowing the full temperature distribution of the machine. This model would have such structure:

$$p(t) = S \cdot T(t) \quad (1)$$

where $p(t)$ is the position variation at a point of interest, $T(t)$ is the temperature field in the machine and S a thermoelastic coupling field.

We propose a reduced order model with the minimum number of temperature inputs (m), which would look like this:

$$p(t) = S_m \cdot T_m(t) \quad (2)$$

Where T_m is a vector with the m measured temperatures and S_m is the thermoelastic coupling vector.

This model is static, since it only considers current temperature values. From experience, it has been observed that more accurate predictions can be obtained in some cases when a dynamic model is used, and therefore $S_m(s)$ is a vector of thermoelastic coupling transfer functions.

$$p(t) = S_m(s) \cdot T_m(t) \quad (3)$$

In general, the thermal error changes within the machine workspace, and therefore the model structure needs to be modified to include a dependence with the machine axis position. A simple and effective way to do this is by a scalar factor K which depends on the position of axis $x_1, x_2, \dots$

$$p(t) = K(x_1, x_2, \dots) \cdot S_m(s) \cdot T_m(t) \quad (4)$$

This approach has led to the development of an accurate and robust thermoelastic model. The simplicity of the mathematical approach helps with robustness, while the accuracy mainly relies on the proper selection of the location of the thermal probes and on the characterization test.

Machine Characterization

Given the number of effects that can affect thermal growth, and the long time required to characterize this effect, it is critical to identify the main error sources that need to be compensated, to focus the effort on these.

For the machines analyzed here, the most relevant heat source is the heat generated by the spindle/quill rotation, in bearings, gears and motor mainly. Therefore, characterization tests consist of measuring machine temperatures and growth while the spindle speed follows certain profiles.

Since each spindle needs to be characterized, the goal was set to characterize each of them in a test that could run during the night without operator presence, and thus not interfering with the machine setup process and minimizing the cost.

In cases where highest precision is required and ambient temperature could have a relevant influence on linear scales, longer tests would be required to allow for higher ambient temperature changes.

Implementation

The implementation of the thermal error compensation system requires the following elements:

- Temperature sensors are embedded in the machine in selected key points and read by the CNC. The sensors are typically located in structural elements and near the heat sources to detect variations as soon as possible.
- Data acquisition: An external computer is used to register these temperatures, and therefore a communication between the PC and the CNC needs to be implemented. It typically changes between manufacturers.
- Data processing: A robust algorithm for model parameter identification needs to be implemented that does not require operator intervention.
- Compensation model: The model with the identified parameters needs to be implemented in the CNC. Again, each CNC manufacturer allows for different implementation methods and thus specific solutions are needed.

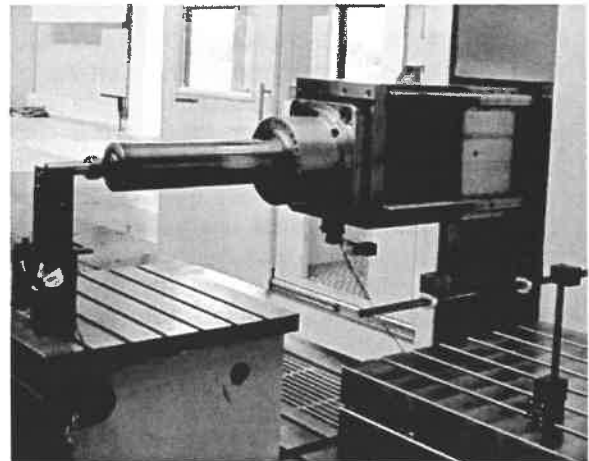


FIGURE 3. Quill TCP measurement in XYZ

EXPERIMENTAL VALIDATION

The effectiveness of the compensation strategy is illustrated next. First the results of the characterization test are presented, and then the validation test results, where the thermal error reduction during a 14h quill test on a SORALUCE machine is shown.

Characterization test

In order to calibrate the compensation model, a characterization test has been performed on the machine. The quill speed profile and the measured TCP position variations are shown next.

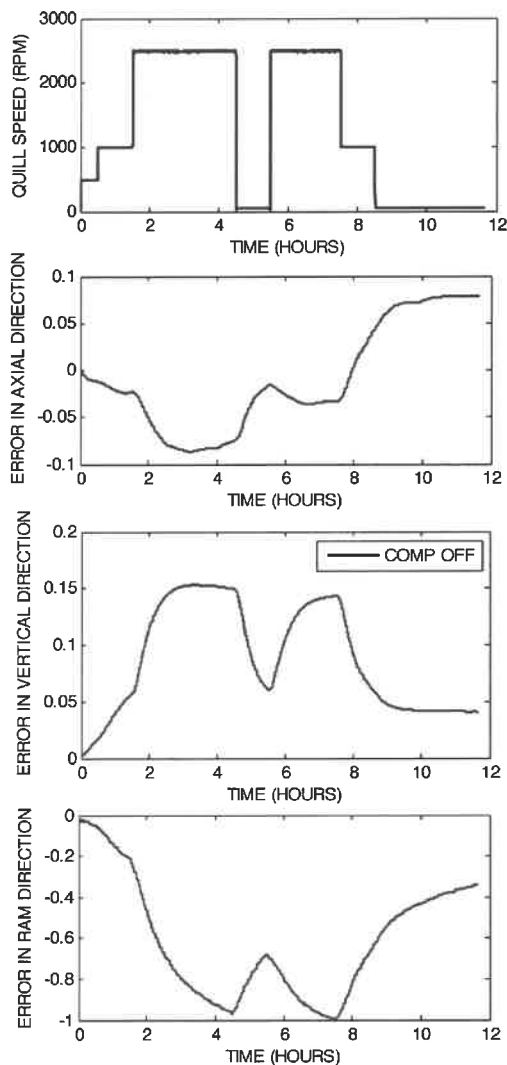


FIGURE 4. Quill TCP error measured during characterization test, normalized to maximum measured deformation

Validation test

Using the information from the calibration test, the compensation model was calculated and implemented in the machine control. A different quill speed profile was used to validate the robustness of the model.

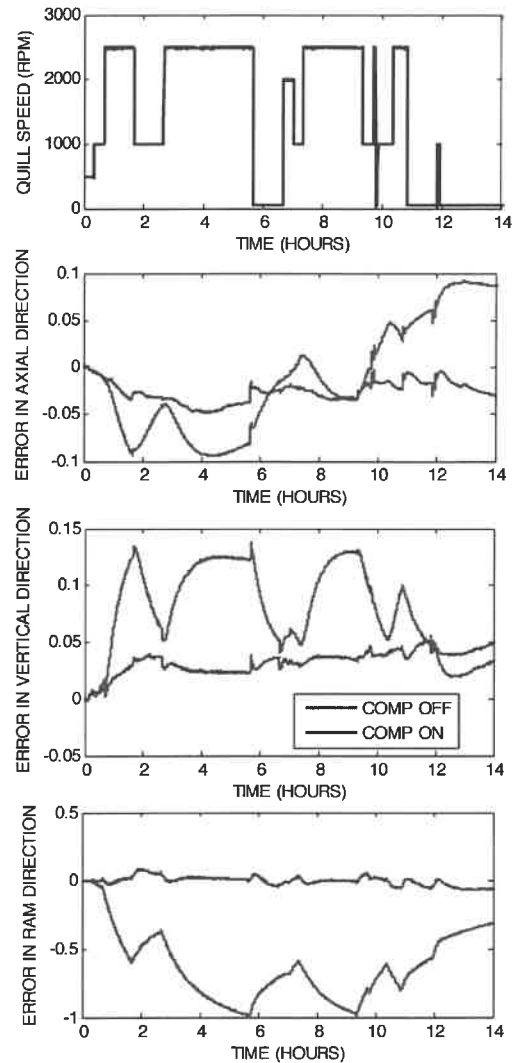


FIGURE 5. Quill TCP error compensation results, normalized to maximum measured deformation

The results show a clear improvement in the TCP accuracy. In X direction, error is reduced to 25% of the machine error, in Y to 40% and in Z to 15%. In absolute displacement, the thermal compensation has reduced the TCP error down to 20% of the original error.

CONCLUSIONS

A thermal error compensation system for large heavy duty milling-boring machines has been presented. The full solution has been developed with industrial feasibility in mind.

The machine characterization tests, thermoelastic modeling and control implementation have been discussed.

The experimental results show a clear improvement in the accuracy of the machine, and thus the validity of the proposed thermal compensation system.

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Development of a System for 3-D Micro Metrology Using an Ultra-Small-Diameter Optical Fiber Probe

- Optical Analysis and Evaluative Experiment Using the 0.4 μm Diameter Stylus Shaft -

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INTRODUCTION

In recent years, demand has increased for a method that can precisely measure microstructures of mechanical micro parts, micro electro mechanical systems (MEMS), micro molds, optical devices, and micro holes, etc. However, it is very difficult to precisely measure the shape of micro structure with a large length-to-diameter (L/D) ratio because of the difficulty of probe fabrication and sensing method with very small measuring force. Studies have reported micro structure measurement techniques employing a variety of probes such as optical probes, vibroscanning probes, vibrating probes, tunneling effect probes, opto-tactile probes, fiber deflection probes, optical trapping probes, and diaphragm probes [1-9].

This paper presents a system for 3-D micro structure measurement using an optical fiber probe that is available as a kind of displacement measuring probe with low contact force and that has a wide measuring range. The stylus of the optical fiber probe was fabricated using an acid-etch technique. The shaft of the stylus is not necessarily rigid in order to detect the measuring force because its deflection is measured by a non-contact method. In this research, the minimum diameter of the available stylus shaft was examined using a finite-difference time-domain (FDTD) method. Then, the stylus shaft with a diameter of 0.4 μm was fabricated by using an acid-etch technique and the characteristic of the stylus shaft in the process of displacement detection was described. Furthermore its performance was evaluated experimentally. As a result, it was shown by an FDTD simulation that the minimum diameter of

the available stylus shaft was about 0.2 μm for a laser diode with a 375 nm wavelength. Also, the stylus shaft with a diameter of 0.4 μm could be manufactured, and the resolution of the measurement system using the stylus shaft with a diameter of 0.4 μm was approximately 5 nm.

MEASUREMENT PRINCIPLE

Figure 1 is a diagram of the optical system. The fiber probe consists of a stylus shaft and a stylus tip. The probing system consists of the fiber stylus, two laser diodes with a 375 nm wavelengths (LDX, LDY), and two dual-element photodiodes (PX, PY) in the X- and Y-directions. The stylus shaft is installed between the laser diodes and the dual-element photodiodes, which are oriented orthogonally. The laser diodes are mounted above the stylus tip, and the focused laser beams are irradiated along the X- and Y-directions onto the stylus shaft. The two dual-

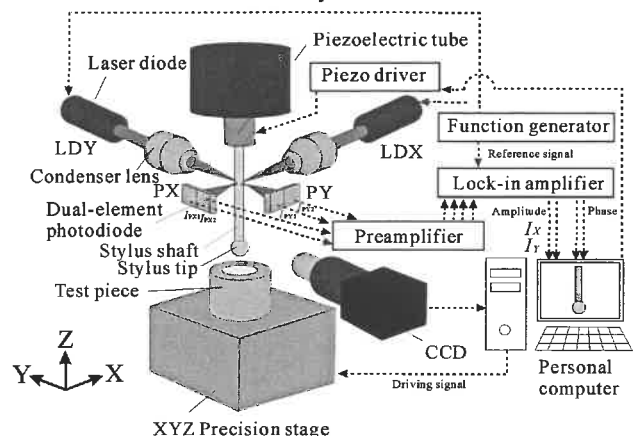


FIGURE 1. Measurement system using optical fiber probe.

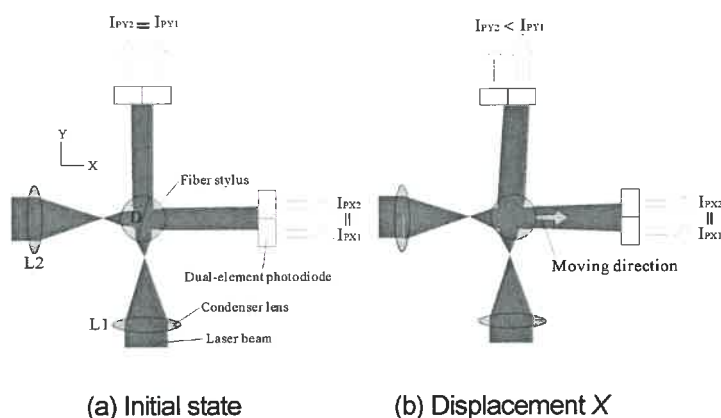


FIGURE 2. Principle of measurement.

element photodiodes are located opposite the laser diodes, beyond the stylus. The laser beams that pass through the stylus shaft are received by these dual-element photodiodes. The beam intensities detected by the photodiodes are converted into voltages and are identified by the symbols I_{PX1} , I_{PX2} , I_{PY1} , and I_{PY2} (V). A charge-coupled device is employed to monitor the positions of the stylus and test piece during the setting up of the equipment and during measurement.

A cross-sectional diagram of the X-Y plane of the optical system shown in Fig.1 is shown in Fig. 2 to illustrate the measurement principle of the optical fiber probe.

Before the stylus tip contacts the measured surface, the light intensity measured by each element of the dual-element photodiode is equal ($I_{PX1} = I_{PX2}$, $I_{PY1} = I_{PY2}$), as shown in Fig. 2(a). When the stylus tip contacts the measured surface in the X-direction, the stylus shaft is displaced, and the light intensity of each element of the dual-element photodiode becomes unequal ($I_{PX1} > I_{PX2}$, $I_{PY1} = I_{PY2}$), as shown in Fig. 2(b). When the stylus shaft is displaced in the +X-direction, the angle of refraction of the laser beam that passes through the stylus shaft in the Y-direction changes due to a shift in the part of the stylus shaft being irradiated. Additionally, when the stylus tip contacts the measured surface in the Y-direction, the light intensity of each element of the dual-element photodiode becomes unequal ($I_{PX1} < I_{PX2}$, $I_{PY1} = I_{PY2}$). As a result, the contact direction and magnitude of the displacement of the stylus tip can be obtained from output signals I_X and I_Y . The noise present in output signals I_X and I_Y is removed by synchronous detection using a lock-in amplifier. The displacement of the stylus is magnified by using it as a rod lens. The surface of the micro-

structure is measured by recording the coordinates at which the stylus makes contact with the surface being measured and the displacement of the stylus.

The output signal I_X in the X direction using I_{PY1} and I_{PY2} and output signal I_Y in the Y direction using I_{PX1} and I_{PX2} are defined by Equations (1) and (2), respectively.

$$I_X = I_{PY1} - I_{PY2} \quad (1)$$

$$I_Y = I_{PX1} - I_{PX2} \quad (2)$$

FABRICATION OF THE STYLUS SHAFT

The stylus shaft was fabricated using an acid etch technique. First, the step index multi-mode optical fibers (core diameter: 100 μm , clad diameter: 125 μm) were stripped of their plastic layers. The tips of the fibers were then immersed in a 25 % hydrofluoric acid solution, and hydrofluoric acid etching was carried out at room temperature (23 $^{\circ}\text{C}$). The diameter of the probe shaft was measured with an optical microscope. Figure 3 shows schematics of the stylus shaft at three different stages of the hydrofluoric acid-etch process. After this process, the stylus shaft was rinsed with water and acetone. Figure 4 shows the experimental results, with the etching time on the horizontal axis and the resultant diameter of the stylus shaft on the vertical axis. Figure 5 shows the photograph of the tip of the stylus shaft. It is confirmed that the diameter of the thinnest part of the stylus shaft is about 0.4 μm . The length of the etching part is about 1500 μm . In future, the fabricated stylus shaft and glass stylus tip of 1 μm diameter will be glued together.

OPTICAL ANALYSIS USING FDTD METHOD

In the case in which the diameter of the stylus shaft is smaller than the diameter of the laser spot, the detection sensitivity decreases because of the reduction in the light intensity that passes through the stylus shaft and the influence of diffraction.

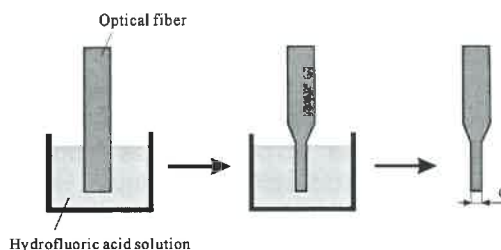


FIGURE 3. Stylus shaft during the hydrofluoric acid etching.

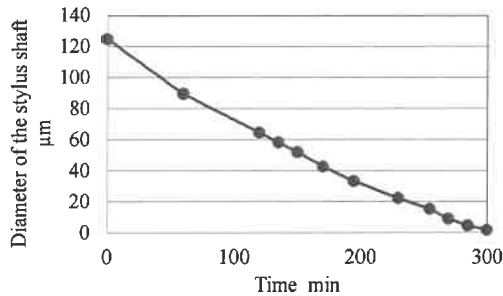


FIGURE 4. Relationship between the etching time and diameter of the stylus shaft.



FIGURE 5. Photograph of the stylus shaft.

Therefore, the minimum diameter of the available stylus shaft was examined using a finite-difference time-domain (FDTD) method. This method solves the Maxwell's equations in the time domain.

In the simulation, the focused laser beams of a Gaussian distribution were irradiated along the Y-direction onto the stylus shaft. Then, the intensities of the transmitted, refracted, and diffracted light detected by the dual-element photodiode were analyzed, and I_x was calculated. The wavelengths of the laser beams used in this simulation were 375 nm and 650 nm. Figure 6 shows the analytical result of the relationship between calculated value of output signal I_x and the diameter of the stylus shaft when the stylus shaft was displaced 0.1 μm in the X-direction. Figure 7 shows the distribution of light intensity around the stylus shaft, generated using the FDTD method.

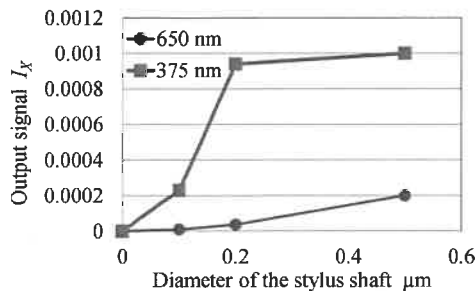


FIGURE 6. Relationship between calculated value of output signal I_x and the diameter of the stylus shaft.

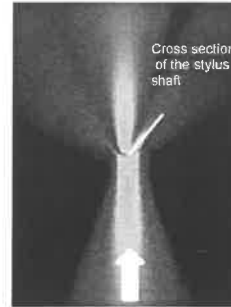


FIGURE 7. Distribution of light intensity around the stylus shaft using the FDTD method.

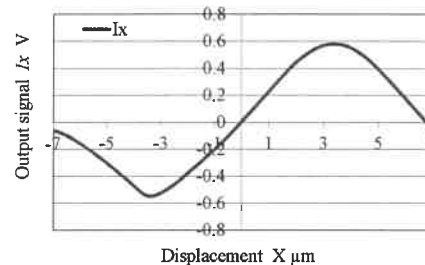


FIGURE 8. Output voltage I_x induced by the X-displacement.

As described in the following section, the measurement resolution of the stylus shaft with a diameter of 0.4 μm was about 5 nm when a laser diode with a 375 nm wavelength was used. The analytical results show that for a laser diode with a 375 nm wavelength, the output signal of the stylus shaft with a diameter of 0.2 μm was almost equal to that with a diameter of 0.5 μm. Therefore, it was shown that the minimum diameter of the available stylus shaft was about 0.2 μm for this laser diode.

STYLUS CHARACTERISTICS

Figure 8 shows the experimental change in output signal I_x when the stylus shaft with a diameter of 0.4 μm was displaced in the $\pm X$ -direction. The horizontal axis shows the displacement of the stylus, and the vertical axis shows the change in output signal I_x . When the stylus shaft is displaced in the X-direction, it can be verified that the fiber probe is available as a displacement sensor because the rate of change in I_x can be approximated as a straight line within a range of ± 3 μm in the X-direction.

MEASUREMENT RESOLUTION

An experiment was carried out to evaluate the measurement resolution of the stylus shaft with a diameter of 0.4 μm in the X direction. The change in output signal I_x was investigated

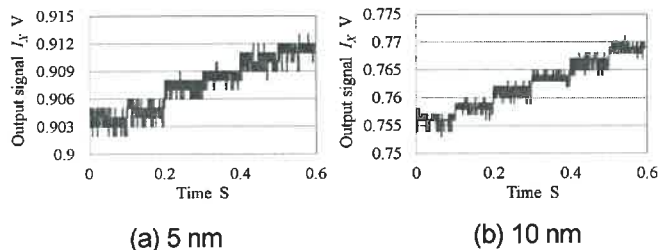


FIGURE 9. Variation in output voltage I_X induced by each step feed of (a) 5 nm and (b) 10 nm.

when the stylus was displaced by 5 and 10 nm/0.1sec steps in the X direction, with the results shown in Fig.9. The horizontal axis shows the measurement time, and the vertical axis shows the change in output signal I_X in the X direction. It is possible to distinguish the 5 nm step, which shows that the resolution of the measurement system using the stylus shaft with a diameter of 0.4 μm is approximately 5 nm.

SUMMARY

This paper has described a system for measuring 3-D micro structures using an optical fiber probe. For this research, the stylus shaft with a diameter of 0.4 μm was fabricated using an acid-etch technique. Further, the minimum diameter of the available stylus shaft was examined using an FDTD method. Finally, the stylus characteristic and the measurement resolution were examined. As a result, it was shown by an FDTD simulation that the minimum diameter of the available stylus shaft was about 0.2 μm for a laser diode with a 375 nm wavelength. In addition, the stylus shaft with a diameter of 0.4 μm could be manufactured, and the resolution of the measurement system using this shaft was found to be approximately 5 nm.

ACKNOWLEDGMENT

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A STUDY ON MATERIAL INFLUENCES IN DIMENSIONAL COMPUTED TOMOGRAPHY

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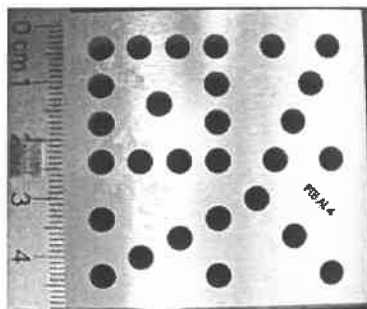


FIGURE 1. Metallic plates with 28 holes drilled through. The large plate is made of aluminum while the small plate is made of steel.

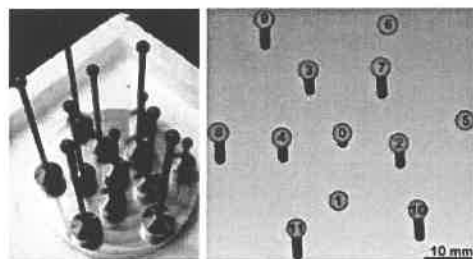


FIGURE 2. 'Styli forest' composed of aluminum oxide (ruby) balls on carbon fiber stems. The balls are numbered in order of increasing 3D distance from the center ball, ball zero.

The main purpose of this study was to determine if there are major additional length measurement errors resulting from additional material surrounding a workpiece when measurements are taken using X-ray computed tomography (CT). This paper presents measurements carried out in January 2014 by the Center for Precision Metrology at the University of North Carolina (UNC) – Charlotte within the ISO TC 213 WG 10 experimental study on material influence in dimensional CT [1, 2]. Two reference objects were used in this study: a hole plate made from aluminum (Al; size 48 mm x 48 mm x 8 mm) and a smaller hole plate made from steel (size 6 mm x 6 mm x 1 mm). Both items were borrowed from Physikalisch-Technische Bundesanstalt (PTB)-

Germany and consisted of square plates with 28 holes geometrically distributed as shown in Fig. 1. A third artifact, a 'styli forest' composed of aluminum oxide (ruby) balls on carbon fiber stems (see Fig. 2) created by the National Institute of Standards and Technology (NIST) [3], was also tested. Measurements in this study were performed using a Zeiss Metrotom 800, a micro-focus X-ray CT machine [4, 5], following the measurement procedures provided in [1, 2]. Dimensional results were calculated from the tomography data with VG Studio Max 2.2 software using the standard adaptive threshold method for segmentation [6].

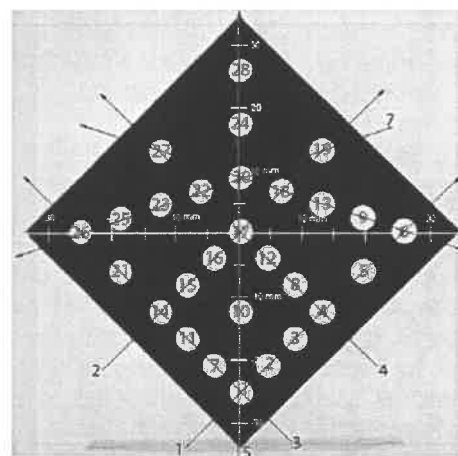


FIGURE 3. Aluminum plate orientation used during the CT scan. The arrows show the directions to set the lengths measurements.

MEASUREMENT STRATEGY

Figure 3 shows the orientation of the aluminum plate in the CT machine. In each direction, the uni-directional distances (center-to-center, between the centers of two holes) and bi-directional distances (point-to-point, between the points of the two holes with the largest distance) are measured. For the evaluation of both kinds of lengths, 11 points illustrated in Fig. 4 are defined on each hole in accordance with the instructions

given in [2]. One example of uni-directional length is the distance between the centers of hole 1 and hole 26, computed as the distance between the points o-1 and o-26, where 'o-k' is the point 'o' (see Fig. 4) on the k^{th} hole. On the other hand, an example of bi-directional length is the distance between the hole 1 and hole 26 computed as the distance between the points h-1 and c-26, where 'c-k' or 'h-k' is the point 'c' or 'h' (see Fig. 4) on the k^{th} hole. Table 1 shows the number associated with each measurand in the uni-/bi-length evaluations and the reference points of the holes to be measured.

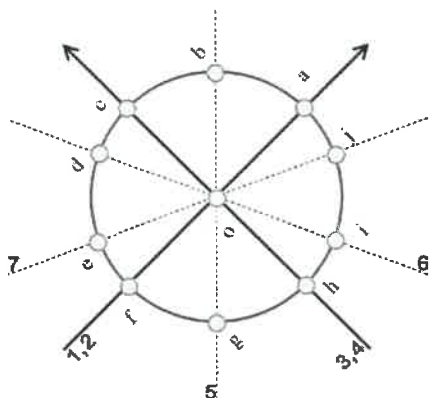


FIGURE 4. Circle of intersection between the mid-plane of the hole plate and a hole with a set of 11 identified points for each hole.

Figure 5 shows the orientation used for scanning the steel hole plate. The nomenclature for the holes and directions along which lengths measurements were taken are identical to the aluminum plate in Fig. 3, despite the change in object's orientation. Because steel is generally known to be a relatively heavier material than aluminum, with bigger X-ray beam attenuation coefficients (density), it is expected to produce more artifacts (distortions) on the CT reconstructed image than aluminum, so a re-orientation of the part was needed to minimize this effect when placing it closer to the X-ray source. This is the rationale for the re-orientation of the steel plate (see Fig. 5) respect to the aluminum plate orientation from Fig. 3. Because the steel plate is 8 times smaller than the aluminum plate, it was possible to place it closer to the X-ray source inside the cone radiation beam such that a magnification close to 8 times bigger could be used (not exactly 8, to avoid locating the sample so close to the source and interfering with it during its rotational movement). Other specific CT parameters used for scanning each part are specified in the results section.

TABLE 1. Labels associated for dimensional evaluations. 1 to 35 correspond to uni-directional (center-to-center) lengths, while 36 to 82 are bi-directional (point-to-point) distances. Reference points are based on Fig. 4. The measurands are ordered by ascendant magnitude but separated between uni-/bi-directional distances.

#	From	To	#	From	To
1	o-14	o-15	42	e-1	j-1
2	o-4	o-8	43	g-1	b-1
3	o-1	o-2	44	f-1	a-1
4	o-1	o-7	45	f-1	a-1
5	o-6	o-9	46	h-1	c-1
6	o-26	o-25	47	h-1	c-1
7	o-28	o-24	48	h-4	c-8
8	o-1	o-3	49	f-14	a-15
9	o-4	o-12	50	f-1	a-2
10	o-14	o-16	51	h-1	c-7
11	o-1	o-11	52	i-6	d-9
12	o-6	o-13	53	e-26	j-25
13	o-26	o-23	54	b-28	g-24
14	o-28	o-20	55	h-4	c-12
15	o-1	o-4	56	f-14	a-16
16	o-4	o-17	57	f-1	a-3
17	o-14	o-17	58	h-1	c-11
18	o-1	o-14	59	i-6	d-13
19	o-6	o-18	60	e-26	j-23
20	o-26	o-22	61	b-28	g-20
21	o-28	o-17	62	h-4	c-12
22	o-6	o-20	63	f-14	a-17
23	o-26	o-20	64	f-1	a-4
24	o-14	o-18	65	h-1	c-14
25	o-4	o-22	66	i-6	d-18
26	o-1	o-5	67	e-26	j-22
27	o-1	o-21	68	b-28	g-17
28	o-14	o-19	69	i-6	d-20
29	o-4	o-27	70	e-26	j-20
30	o-1	o-6	71	h-4	c-22
31	o-1	o-26	72	f-14	a-18
32	o-28	o-10	73	f-1	a-5
33	o-6	o-27	74	h-1	c-21
34	o-26	o-19	75	h-4	c-27
35	o-28	o-1	76	f-14	a-19
36	h-4	c-4	77	f-1	a-6
37	f-14	a-14	78	h-1	c-26
38	d-6	i-6	79	b-28	g-10
39	e-26	j-26	80	i-6	d-27
40	b-28	g-28	81	e-26	j-19
41	i-1	d-1	82	b-28	g-1

For the third artifact, the 'styli forest' from Fig. 2, uni-directional lengths were measured (distances center-to-center between the spheres), but not

point-to-point (bi-directional) distances; measurements of form (sphericities) as well as average diameters sizes for each sphere were taken instead.

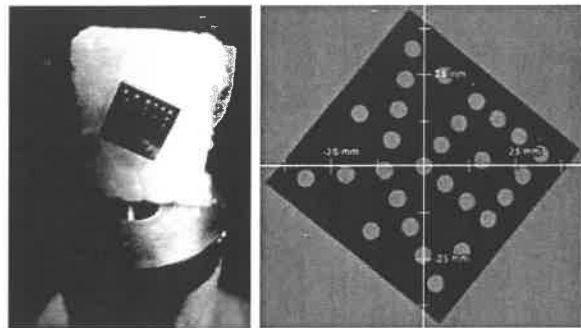


FIGURE 5. Steel plate orientation used for the CT scan. The nomenclature for the holes and directions for lengths measurements are identical as in Fig. 3.

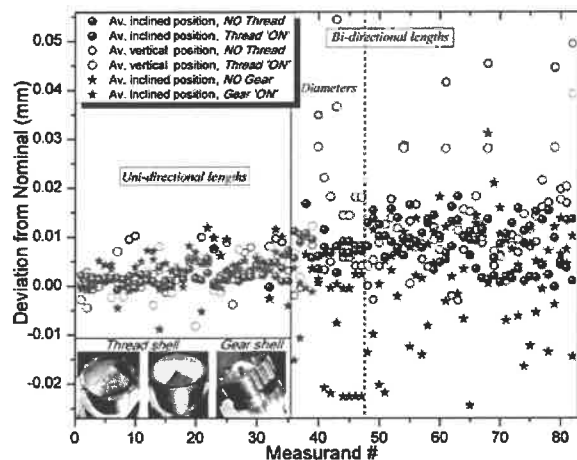


FIGURE 6. Length measurements for the aluminum hole plate as deviations from the nominal values claimed in reference [2]. The measurements labeled with 'NO' are taken without covering the plate with a shell during the CT scan, while those labeled with 'ON' were obtained by scanning the part inside the indicated aluminum shell. Positioning of the part is the same between 'NO' and 'ON' configuration when referred to the same shell type. The x-axis (Measurand #) numbers are those identified in Table 1.

To investigate material effects, the reference objects were covered during the CT scans with aluminum shields designed to present the specimen with a varying thickness of aluminum material (see photographs on Fig. 6, 7, and 10). Two shields provided by NIST were designed to be highly irregular with respect to the amount of

material they present to the X-ray beam when scanning the aluminum hole plate and the styli forest, and a smaller shield was manufactured at UNC Charlotte to be used with the small steel hole plate. Since the plates are not calibrated, this study is investigating differences between length measurements from two different scans, a scan run of the reference part only (without the shell covers) and the second scan run with the shells added during the CT data collection.

TABLE 2. Machine parameters used during the CT scans for the measurement configurations of Fig. 6.

Aluminum Hole Plate	Thread shell		Gear shell	
	'NO'	'ON'	'NO'	'ON'
V (kV)	100	130	100	130
I (uA)	161	270	161	300
It (s)	267	267	400	400
SOD (mm)	300	300	300	300
SDD (mm)	800	800	800	800
Pre-filter	Al 0.5	Cu 0.25	Cu 1.0	Cu 1.25
# of IA	5	10	10	10
Spot (um)	40	40	40	40
Vx (um)	50	50	50	50
M	2.56	2.56	2.56	2.56
projections	800	800	800	800

RESULTS

Figure 6 shows the measurement results for the aluminum plate at three configurations with two aluminum covers indicated by photographs: the 'Thread' and the 'Gear' shells (left bottom pics in Fig. 6). To establish consistency, each configuration was measured more than once, preferably taking two 'NO-shell' condition measurements before the 'shell-ON' measurements and two more after. Although the inclination of the part was different between measurements taken with the 'Thread' and 'Gear', the part stays in the same position in each case while taking 'NO shell' conditions measurements at each configuration, this becoming a reference for comparison. During measurement runs with the part covered by a shell, the X-ray's energy was generally increased to enhance contrast of the CT image and penetrate the barriers interposed by the cover shells. To optimize contrast and minimize overall path length for the X-rays (avoiding beam hardening effects), an inclined position for the part is preferred, but inclinations may vary between different parts and/or shells. The scan parameters used during each run are shown on Table 2, and the nomenclature parameters are as

follows: V = voltage, I = current, It = integration time, SOD = source-object-distance, SDD = source-detector-distance, IA = image averaging, Vx = voxel size, and M = magnification. Similar procedures were followed to obtain the CT scans on the small steel plate in the configurations with and without the aluminum shell for the orientation pictured in Fig. 7. Also, the 'styli forest' with the aluminum oxide (ruby) balls was tested in this manner by using the 'Thread' and 'Gear' shells, see Fig. 8.

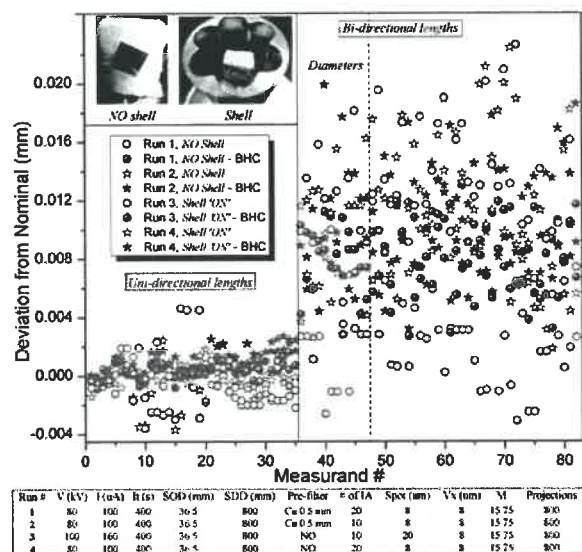


FIGURE 7. Length measurements for the steel hole plate as deviations from the nominal values claimed in reference [2]. BHC indicates when beam hardening correction is applied to the CT data. 'NO' and 'ON' labels follow the same convention stated in Fig. 6. The x-axis (Measurand #) numbers are referred in Table 1.

From Fig. 6, we see there is not a noticeable deviation between length measurements taken for the aluminum plate in 'shell-ON' condition in respect to 'NO-shell' configuration, other than a natural dispersion expected from the experiment. This leads to the conclusion that there is not an influence on the dimensional measurements for the aluminum hole plate when it is surrounded by additional material, like the aluminum cover shell. Only by increasing the power of the X-ray tube is it possible for the X-rays to penetrate the interposed aluminum barrier and transmit through the part to hit the detector without altering the dimensional measurements taken on the hole plate or object of study. The data dispersion in Fig. 7 is higher for the bi-directional lengths. This is likely because the distances are being taken between two individual points, rather than the

hole centers calculated from many points, making the result more sensitive to measurement deviations. Locating positions using single points from the CT data can bring discrepancies; averaging neighbor locations in the point cloud around the desired position may increase the precision. Also, it is not easy to precisely locate the part (edges) boundaries in the surface determination phase of the CT data analysis. For the case of circular surfaces, boundary location errors generally do not affect the center locations since errors are expected to expand homogeneously in all directions and deviations will average out, and thus the center-to-center distances do not get affected. The resolution and repeatability of surface determination in CT generally depends on the voxel size selected for the scans, which was 50 μm for the aluminum hole plate. This gives us an expected dispersion more or less between +25 μm and -25 μm for dimensional measurements in Fig. 6.

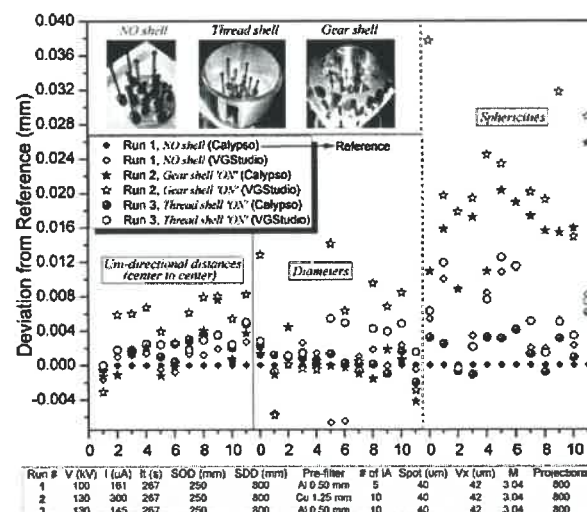


FIGURE 8. Length measurements for the 'styli forest' balls as deviations from data computed with Calypso in 'NO shell' condition. 'NO' and 'ON' labels follow the same convention stated in Fig. 6, but the software used for the analysis is labeled in parenthesis. The x-axis numbers determine the ball on which the measurand is taken recalling Fig. 2. Uni-directional lengths are measured between the referred ball (number) center and the zeroth sphere's center.

The previous analysis can be extended for the steel hole plate. However, because steel is a heavier material than aluminum, it was expected to produce more artifacts on the reconstructed volume as a result of hardening effects [8]. Therefore, beam hardening correction (BHC)

algorithms were applied. Both, 'BHC' and 'No BHC' results are plotted in Fig. 7. Although errors from beam hardening were expected, for our tests there is not much difference between the 'BHC' and 'No BHC' results. The dispersion for the uni-directional lengths lies between $-4\ \mu\text{m}$ and $+4\ \mu\text{m}$, as expected for a machine with a maximum permissible error for circle point location stated as $\text{MPE}=(4+L/100)\ \mu\text{m}$ with L given in mm. Bi-directional measurements show bigger dispersion than for the uni-directional ones, for the same reasons explained in the case of the aluminum plate. The CT data in this case (steel plate) was taken with $8\ \mu\text{m}$ voxel size, and the data dispersion for the bi-directional distances lies mostly between 0 and $16\ \mu\text{m}$ (see Fig. 7), which may lead to conclude the bidirectional deviations are shifted $8\ \mu\text{m}$ above the zero deviation. These results are interesting, but since the hole plates were not calibrated, this cannot be confirmed without measuring the part with a CMM (or other reference instrument) for comparison.

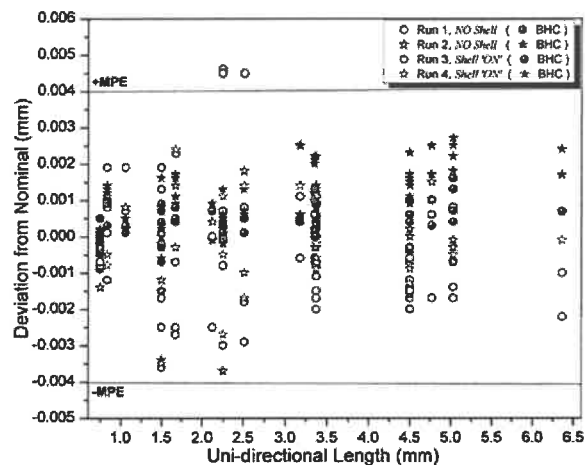


FIGURE 9. Uni-directional (center-to-center) length measurement deviations for the steel hole plate. This is a zoom view of the uni-directional section of Fig. 7, but in a linear scale with the x-axis as a length dimension. Deviations are calculated from nominal values only and MPE represents the maximum permissible error for circle point location stated by the CT manufacturer.

The dimensional measurements reported in Fig. 6 and 7 were computed with VGStudio MAX 2.2. A recalculation with Calypso, a software provided by the manufacturer of the CT equipment used, may produce interesting results for comparison in the future when a complete set of CMM references values for the hole plate measurands

becomes available. For now we have computed differences between results calculated with the two softwares for the ruby spheres in the 'stylus forest' artifact shown in Fig. 2. The configurations using the aluminum shells is the same as described above, 'NO-shell' and 'shell-ON', and results are plotted in Fig. 8. No bi-directional (point-to-point) distances were measured, but measurements of form and sphericity, were obtained. As expected, the possible errors resulting from surface determination or imprecise location of edges is reflected in the dispersion of sphericity values (measurement of form). Length deviations between different measurement runs remain more or less within the MPE limits for the CT equipment used. Hence there does not appear to be a significant difference between dimensional measurements computed with the two softwares, VGStudio and Calypso. In future research, a comparison with values from a CMM will be added to this analysis.

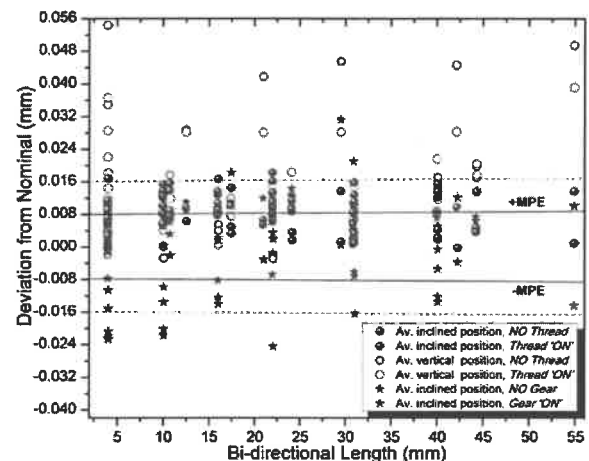


FIGURE 10. Bi-directional (point-to-point) length measurement deviations for the aluminum hole plate. This is a zoom view of the bi-directional section of Fig. 6 in a linear scale. MPE represents here the maximum permissible error for length measurement, as specified for the instrument.

CONCLUSIONS

Because dimensional measurements from a measuring machine other than CT are not available yet, it is not possible to assess the length measurement errors. It is possible to compare the difference between measuring the hole plates with extra-material (aluminum shells) compared with CT measurements taken on scans of the hole plate with no shell. From our results, no noticeable differences are observed for the measurements taken with extra-material (aluminum shell 'ON') around any of the hole

plates studied. There exists experimental data dispersion as expected from the physical process of measurement. Fig. 9 for example, shows a zoom view of the uni-directional lengths in a linear scale for the case of the steel hole plate, and it is worth noting that the dispersion or deviations between different measurement runs remains within the limits of the maximum permissible error for circle center point location stated by the CT manufacturer ($MPE=4+L/100\text{ }\mu\text{m}$ with L given in mm). Similar behavior is repeated by the uni-directional distances in the case of the aluminum hole plate. In contrast, bi-directional measurements display higher dispersion, as can be seen for example in Fig. 10; deviations for the point-to-point measurements on the aluminum hole plate are lying in a region twice wider than the MPE, being MPE in this case the maximum permissible length error, $MPE=(8+L/100)\text{ }\mu\text{m}$ with L in mm. This is to be expected for a bi-directional strategy of measurement limited by two single points to assess (point-to-point) distances, mainly because of possible errors resulting in the edges (surface) determination. Locating very precisely the boundaries during the CT surface determination phase of the CT data analysis is challenging, but averaging neighbor locations in the point cloud around the claimed position to define the point might increase the precision of the bi-directional measurements. Furthermore, it is also noted that in both comparisons (uni-/bi-directional) longer length measurements result in larger deviations from nominal values, which are the result not only of propagation of error with distance but of Feldkamp's cone-beam CT reconstruction algorithm errors when using circular scanning paths [9, 10, 11]. When the calibration data for the items becomes available, this analysis will be revised to include a comparison of these techniques relative to this reference data. As a last observation, the CT measurements of this study were taken with a METROTOM 800, which is primarily used for scanning injection molding industrial parts (e.g., plastics, composite materials, ceramics, and lighter metals). However, steel parts require higher power levels of the source [12]. Therefore, measurements of the hole plates was performed to see the effects of the enclosures (aluminum shells) on the measurement and not the accuracy of the measurement. Authors are planning to repeat this study using more powerful CT system that would ensure more optimal measurement results.

ACKNOWLEDGMENTS

The authors would like to express their appreciation to Markus Bartscher from PTB (Germany) and Vincent D. Lee from NIST (Gaithersburg, USA) for their cooperation in providing the CT artifacts for this study. Also to Carl Zeiss Industrial Metrology, LLC for providing the CT measuring machine and Calypso software. To Volume Graphics GmbH as well, for providing VGStudio MAX. This study could not have been completed without their collaboration.

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Study on dynamic observation of sub-50 nm sized particles in water using evanescent field with a compact and mobile apparatus

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INTRODUCTION

This study proposes the concept and a state-of-art compact and mobile apparatus for dynamically observing sub 50 nm-sized moving particles by using an evanescent field and applies it for several in-situ investigations.

EVANESCENT FIELD

Total reflection occurs when light irradiates the interface from a high refractive index n_1 material, into a lower refractive index n_2 material if the incident angle i is larger than the critical angle, $\arcsin(n_1/n_2)$.

On the reflecting surface in the lower refractive index side, a localized light called an evanescent field that does not propagate in free space is generated and is localized near the surface as shown in Figure 1. The evanescent field intensity decays exponentially with distance from the reflecting surface as shown in the following equation (1). Here z is the distance from the reflecting surface, λ is the incident laser wavelength and I_0 is the incident laser intensity [1, 2].

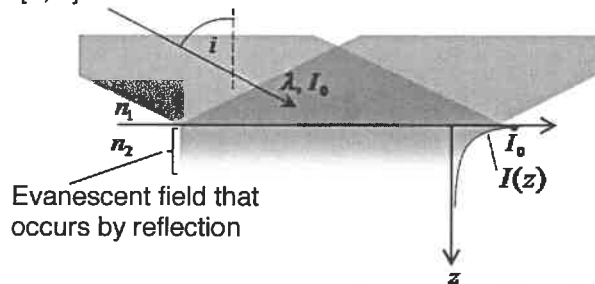


FIGURE 1. Evanescent field at a reflecting surface.

$$I(z) = I_0 \times \exp\left(-\frac{4\pi z \sqrt{n_1^2 \sin^2 i - n_2^2}}{\lambda}\right) \quad \dots(1)$$

LIGHT SCATTERING FROM A PARTICLE NEAR THE REFERENCE SURFACE

Figure 2 shows how nano-sized particles convert an evanescent field to propagating light which is observable by a microscopy system when the particles enter the localized evanescent field. For an example, a particle moves near to the reference surface from left to right direction in the evanescent field, the propagating light intensity becomes maximum when the particle is nearest to the reference surface.

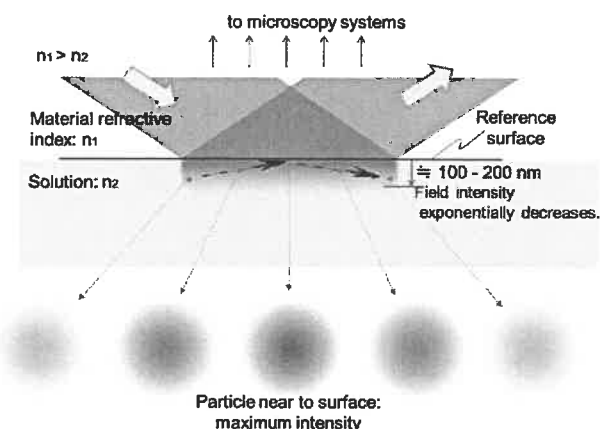


FIGURE 2. Dynamic observation of nano-sized particles in evanescent field.

By this method, only the particles near the reference surface can be visualized and investigated due to the limited range of the localized evanescent field. Therefore, an observed image of the scattered light from each particle, not an image of the particle, will have high contrast since there is no scattered light

from particles that are out of the evanescent field.

DEVELOPED MOBILE APPARATUS

The optical system that was employed in this paper is shown in Figure 3. The scattering propagating light is accumulated to an area sensor of a digital camera with an objective lens (numerical aperture: NA = 0.45) and an infinity-corrected tube lens (focal length: $f = 200$ mm). The tube lens is applied for adjustability of working distance in observation.

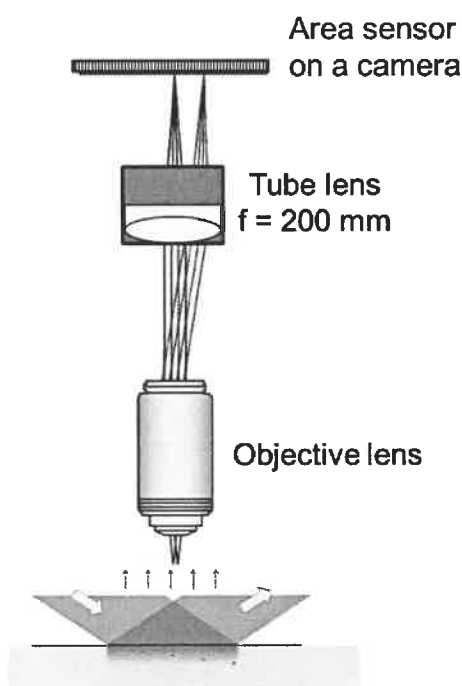


FIGURE 3. Optical system of the developed apparatus.

Figure 4 and Table 1 show the developed experimental apparatus with dimensions of $W \times L \times H \approx 100 \times 150 \times 350$ mm³ and a weight of 15.7 N that could be used for several applications of nano-sized particle in-situ investigations. The pixel resolution of the dynamically observed images was 79 nm/pixel in the following experimentations in this paper. And the n_1 of the reference surface and the n_2 of fluid (water) were 1.52 and 1.33, respectively.

TABLE 1. Developed apparatus specifications.

Laser Diode		
- Wavelength λ	nm	532
- Power	mW	15
- Incident angle i	deg	80
CMOS Camera		
- Pixel size	μm^2	5.5
Optical lens system		
- Total Magnification		70x
- Numerical aperture		0.45

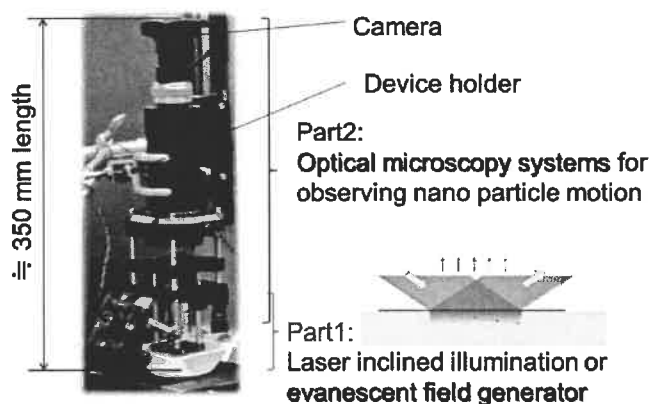
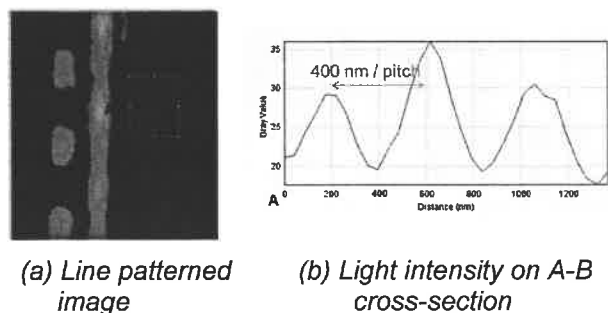


FIGURE 4. Experimental apparatus.

Resolving power of optical microscopy system

An already-known line-space patterned wafer surface was observed to approve the resolving power of the developed optical microscopy system. Figure 5 (a) shows an observed image of a line pattern on the wafer. The light intensity, i.e. grey scale, from the profile A-B in Figure5(a) is shown in Figure5(b). The optical microscopy system was evidently able to distinguish 400 nm separated pattern. This implies the clearly resolving power in the developed apparatus is 400 nm approximately [4].



(a) Line patterned image

(b) Light intensity on A-B cross-section

FIGURE 5. Observation of a test patterned wafer to approve the resolving power

Background subtraction for enhancing the scattering light from moving particles

In practical, there usually are few contaminants on the reference surface in observation. Therefore, scattering light from motionless contaminants, was assumed as a no-change background, which can be obtained by averaging each image frame that was recorded at each corresponding time $t_1, t_2, \dots t_n$ and so on. Then, the scattering light from only moving particles of each image at t_n was enhanced by subtracting the no-change background out as shown in Figure 6. The observation images to be shown in the following experimental results were already subtracted the no-change background.

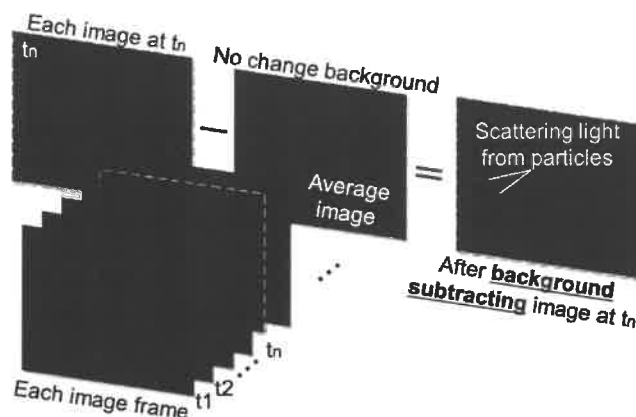
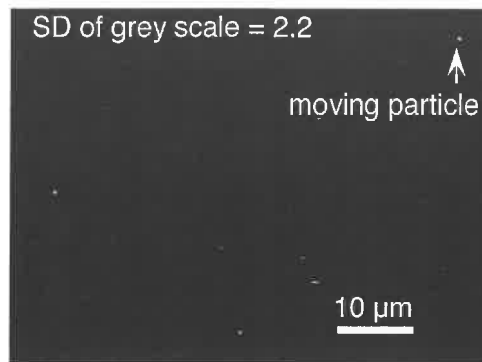


FIGURE 6. Background subtraction

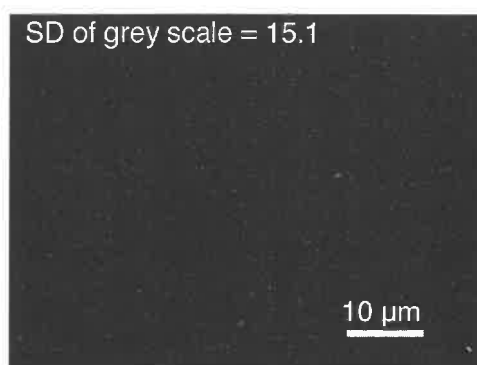
Comparison between observation by evanescent field and bright field observation on 50 nm silica particles

A 50 nm silica particles observed image in polishing slurry in the evanescent field has considerably high contrast (the standard deviation: SD of grey scale was 2.2), comparatively in the usually-used bright field (the standard deviation: SD of grey scale was 15.1) as shown in Figure 7(a) and (b), respectively.

In the evanescent field, there was no scattering light from the particles that were out of the field, which was in the focal plane of optical system. In contrary, there was a lot of blurred scattering light from the particles that were out of the focal plane.



(a) By evanescent field after background subtraction



(b) By bright field after background subtraction

FIGURE 7. Difference in 50 nm silica particles of one observed image

STANDARD Au-PARTICLES OBSERVATION

Standard gold nanoparticles (diameter of 15 ± 2 , 20 ± 2 , 30 ± 2 , 50 ± 3 nm; in water with 0.1mM or 0.1×10^{-3} mol/L of phosphate buffered saline; gold component less than 0.1%w/v or 0.1g/100mL) were observed at 100 frame/s. Scanning electron microscopy images of Au-nanoparticles, diameter of 30 nm and 15 nm are shown in Figure 8 (a) and (b), respectively for examples.

Figure 9 shows a typical observation for moving 30 nm particles. As the results, the 30 nm particles were tracked at 100 frames/s using 2000 μ s exposure for frame

Characteristic of Scattering Intensity with Exposure Time

Maximum scattering intensities that can result from particles moving nearest to the reference surface were plotted in Figure 10 for different exposure times. This characteristic maxima in

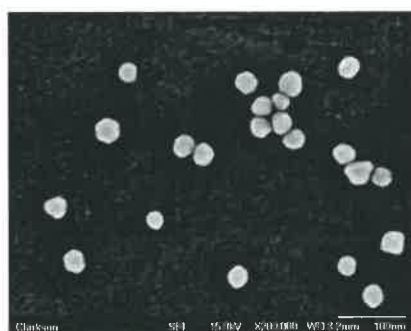
the scattering light intensity can be used as a calibration to determine particle size by proper exposure time.

Observation of Particles under a 4H-SiC Wafer

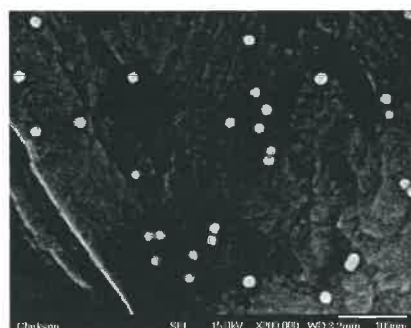
The particles near a SiC surface were also dynamically observed with birefringence in the scattering distribution due to the crystalline substance as shown in Figure 11. This method could also be applied for in-situ investigations of particle behavior [3] on a SiC wafer surface.

CONCLUSIONS

By using the developed compact and mobile experimental apparatus, sub 50 nm-sized moving particles can be dynamically observed. The characteristic peak in intensities of scattered light from particles in an evanescent field can be used to determine particle size with proper exposure time. Not only particles under a reference silica surface, but also those under a 4H-SiC wafer could be observed.



(a) 30 nm particles



(b) 15 nm particles

FIGURE 8: Scanning Electron Microscopy images of the particle to be observed

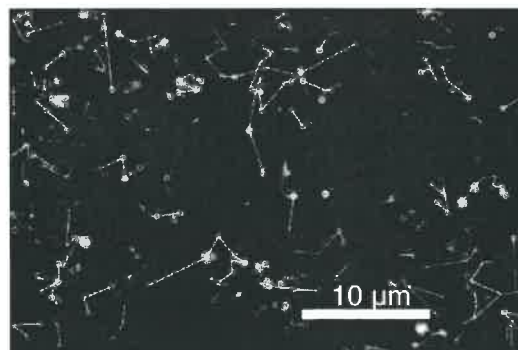


FIGURE 9. Tracking 30 nm particles in 1 s at 2000 μ s of exposure time in the evanescent field.

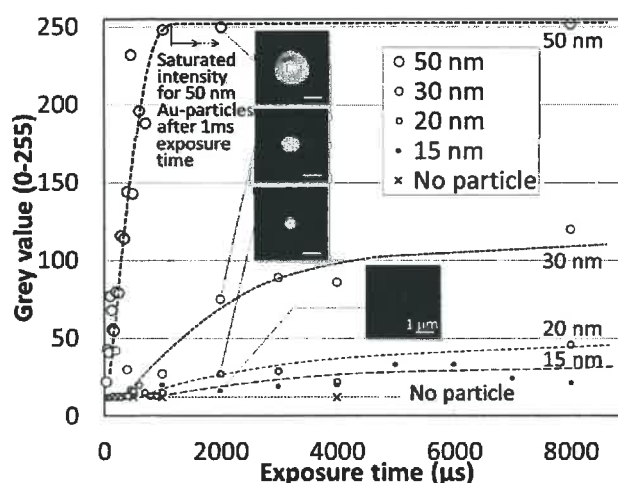


FIGURE 10. Maximum intensity values from 500 frames corresponding to exposure time.



(a) on silica substrate (b) on 4H-SiC substrate

FIGURE 11. Scattering light with birefringence from a 50 nm moving particle

ACKNOWLEDGEMENTS

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INSTALLATION OF A VIBRATING CONTACTING PROBE ONTO AN ULTRA PRECISION CMM

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ABSTRACT

The initial results associated with the installation of the vibrating micro-probe from the National Physical Laboratory (NPL) onto the "Isara 400" 3D coordinate measuring machine (CMM) from IBS Precision Engineering are presented. The mechanical and electrical interface is described. The software used to operate the probing system, and to calculate the deflection and overtravel of the probe, is also described. The probing-point repeatability results are presented, as well as results from measurements of a flat gauging surface.

INTRODUCTION

A micro-CMM probe has been developed at the National Physical Laboratory (NPL), UK [1]. The probe is designed to operate in a vibrating mode to address the effects of the surface interaction forces prevalent at the micrometre scale. These surface interaction forces affect micro-CMM probes through surface adhesion and snap-in to, or, snap-back from, the measurement surface [2].

The vibrating micro-probe consists of a triskelion flexure array, housed in a MEMS device, assembled with a sphere-tipped micro-stylus [3]. The system was designed to vibrate by using six piezoelectric actuators (two on each flexure).

The vibration of the probe is controlled so that the acceleration of the stylus tip is sufficient to counteract the surface interaction forces between the stylus tip and the measurement surface during contact measurement [4] [5].

Interaction with the measurement surface produces a change in vibration characteristics which is determined by two PZT sensors on either end of each flexure.

An image of the NPL vibrating micro-CMM probe is shown in figure 1. The development and laboratory-based verification measurements of the vibrating probe are reported elsewhere [1] [6] [7].

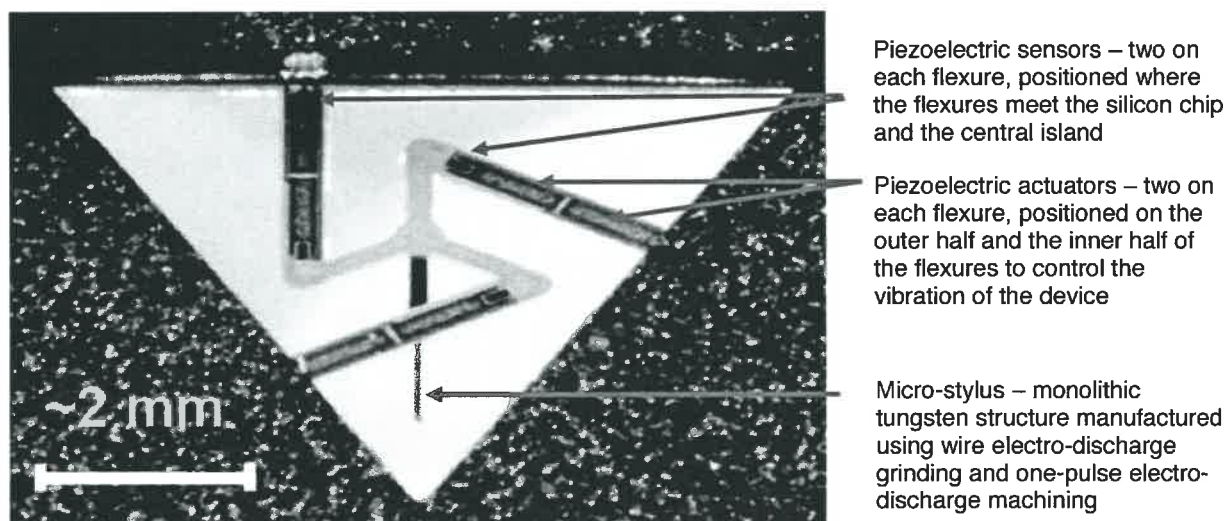


FIGURE 1. The NPL vibrating micro-CMM probe.

Previous work conducted in the area of micro-co-ordinate metrology has concluded that further reduction of the errors associated with measurements taken using micro-CMMs is limited, primarily, by the probing systems [8]. Therefore, it is critical that new systems are developed.

INSTALLATION

Following the successful testing of the vibrating probe in laboratory conditions, the next stage was to install the probe onto a CMM of appropriate precision.

The Isara 400 ultra precision coordinate measuring machine (CMM) from IBS Precision Engineering offers a theoretical measuring uncertainty of approximately 57 nm ($k = 2$) per individual measuring axis over a measuring volume of 400 mm by 400 mm by 100 mm. The measurement of a flat mirror artefact shows a measurement uncertainty of 11 nm ($k = 2$) [9]. An image of the Isara 400 CMM is shown in figure 2.

The measuring principle of the Isara 400 CMM conforms to the Abbe principle in full 3D [10]. The axis positions are measured using laser interferometers, which are aligned to the common "Abbe point" [11].

The Isara 400 was chosen as it makes use of laser interferometers on its movement and measurement axes. Therefore, the measurements taken with the vibrating probe while installed on the Isara 400 can be compared with traceable displacements.

To complete installation, three main areas were considered: the physical interface, the electrical

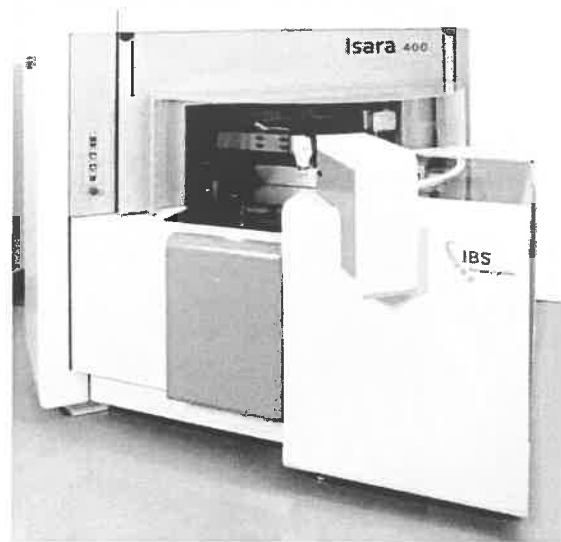


FIGURE 2. The Isara 400 from IBS Precision Engineering

interface and the control and sensing software. To complete the physical interface between the vibrating probe and the Isara 400, a prototype CMM head was designed to allow for easy exchange of probes. The CMM head was also required to provide the function of electrical interface between the probe, the CMM and the control and sensing electronics. To realise the electrical connection, a set of suitable sprung-pin electrical connectors were installed into the CMM head. Electrical connection to the external control and sensing electronics (a set of simple low-pass filters, 2 kHz cutoff) was made using shielded, twisted pair connections.

A schema of the interface is shown in figure 3 and an image of the realised CMM head is shown in figure 4.

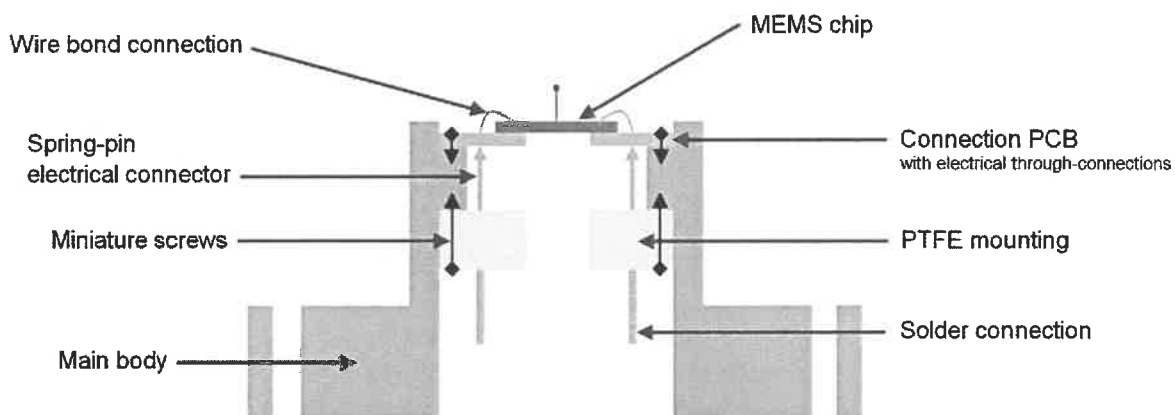


FIGURE 3. Schema of the physical and electrical interface for the vibrating micro-probe.

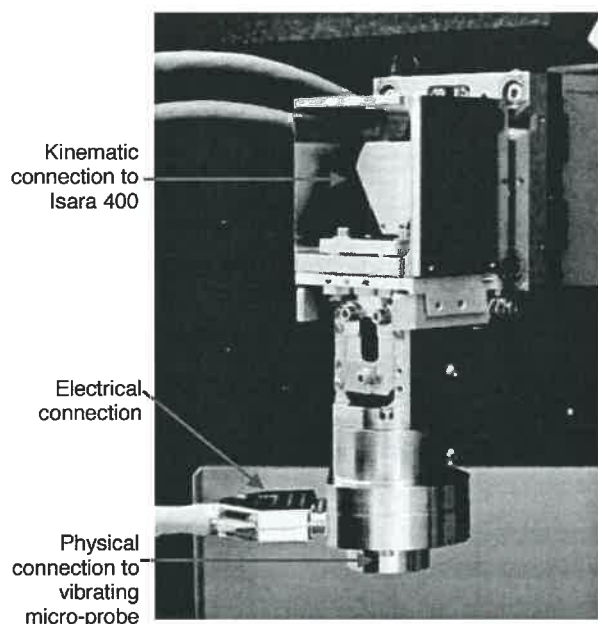


FIGURE 4. The prototype CMM head

The software for the operation of the probe, developed by NPL using LabVIEW from National Instruments, fulfils two main tasks; the control of the vibration of the probe, and the calculation of the probing signal from the sensor output signals. The vibration of the probe is controlled according to a predefined mechanical model of the probe. The probing signals are calculated through Fourier analysis, and are compared with

predetermined calibration information to define overtravel. This process is described in detail elsewhere [7].

Along with the calculation and output of the sensor signals, the operation software also interfaces with the Isara 400 to communicate the current state of the CMM and the probe. This communication includes: distinguishing between the CMM being in motion and stationary with the CMM preparing to probe, probing and an emergency halt signal.

RESULTS

Initial tests involved investigating the probing-point repeatability of the system. To complete this test, a single point was repeatedly contacted approximately one thousand times, and the calculated overtravel of the probe was compared to the reading from the laser interferometer scale on the z-axis of the Isara 400.

The comparison of the signals from the probe and the CMM introduces some error into the measurement as the signals are not synchronous and the CMM is never completely stationary (at the nanometer level). The combination of this asynchronous error, and other errors associated with the calibration and operation of the probe and its interface with the CMM, are all included in the probing-point repeatability. The raw data from this experiment is shown in figure 5.

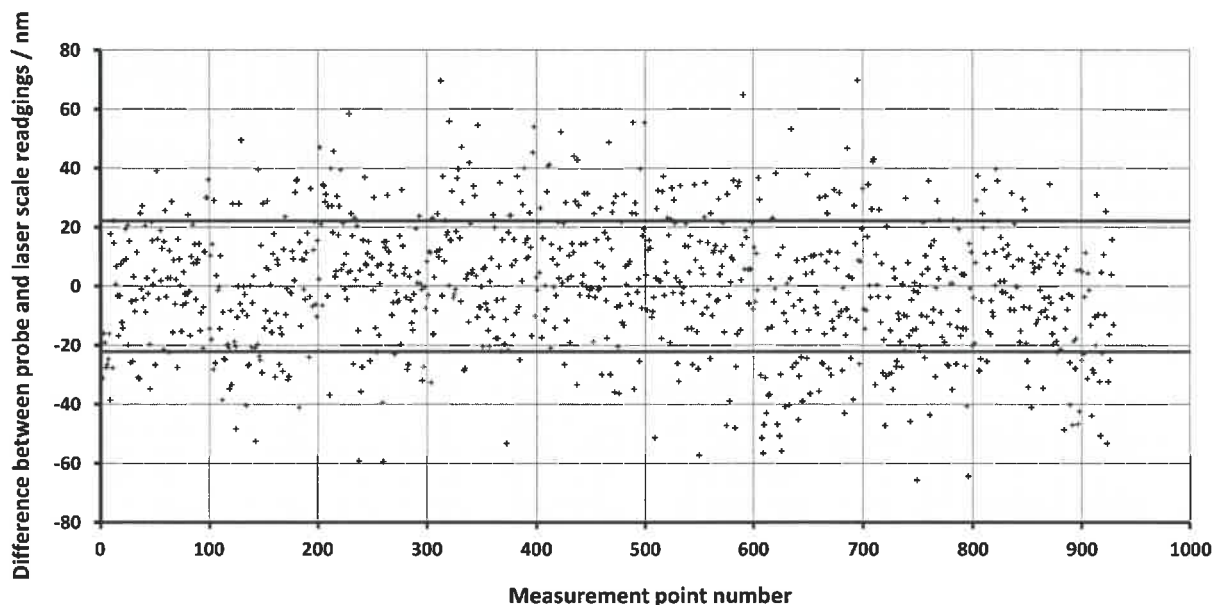


FIGURE 5. Graph showing the calculated difference between the probe overtravel and laser scale reading for a single point measurement, repeated approximately one thousand times. This experiment ran for approximately three hours. The horizontal lines indicate ± 23 nm (one standard deviation).

Results indicate that the repeatability is 23 nm (1σ) over approximately one thousand measurement points. The full range of the deviation of the measured data points is 136 nm. Combined with a calibration uncertainty of the vibrating probe of 14 nm ($k=1$) [7], an initial single-axis probing error of 27 nm ($k=1$) is estimated.

During completion of the one thousand point repeatability test, the environmental temperature of the laboratory changed by less than 0.05 K. It was, therefore, assumed that this effect on the measurements, especially the finite length of the stylus, was negligible. During completion of the one thousand point repeatability test, the position of the x and y axes were also recorded. These lateral axes experience a servo-error of less than 100 nm, but do experience two abrupt shifts of less than $6\ \mu\text{m}$, corresponding with the manual reset of the reference signal of the probe – as described later. These abrupt shifts do not correlate with any discernable shifts in the z -axis. Therefore, it is assumed that any servo-errors can be neglected.

Following the completion of the repeatability tests, two simple geometrical measurements were also completed, using the gauging surface of a grade 0 gauge block as a measurement sample [12]. Data sets consisting of one hundred measurement points, evenly spaced over two separate areas of 5 mm by 5 mm, were collected. Graphical representations of these data (leveled through rotation to a least-squares best fit plane) are presented in figure 6.

The presented data shows deviations from perfect flatness of 137 nm and 152 nm (peak-to-valley) respectively. This deviation is larger than the specified flatness deviation for a grade 0 gauge block (100 nm full field, assuming the gauge meets specifications), but comparable to the full range of the deviation of the measured data points due to repeatability. When including the associated estimated single axis probing error, the results become comparable, however, over the relatively small measured area (compared to the full surface area of the gauging face of the grade 0 gauge block) there is still a larger deviation in measured flatness than would be expected.

DISCUSSION

While completing the described tests, several further error sources became apparent. The effect of some of these error sources will now be discussed. The bespoke operating software developed by IBS Precision Engineering for the Isara 400 operates at 10 kHz. However, the amount of data required to adequately calculate probe signal through Fourier transformation results in a reduced output frequency of probe signals. Therefore, the calculation of the probe signals during operation is limited to an output frequency of 10 Hz. This limits the ability of the probe to detect any small (nanometre level) high frequency movements of the Isara 400 during calculation of the measurement point.

High precision probe signal calculations could not be realized due to the electrical characteristics of the probe coupled with the low-pass filters used to condition the sensor

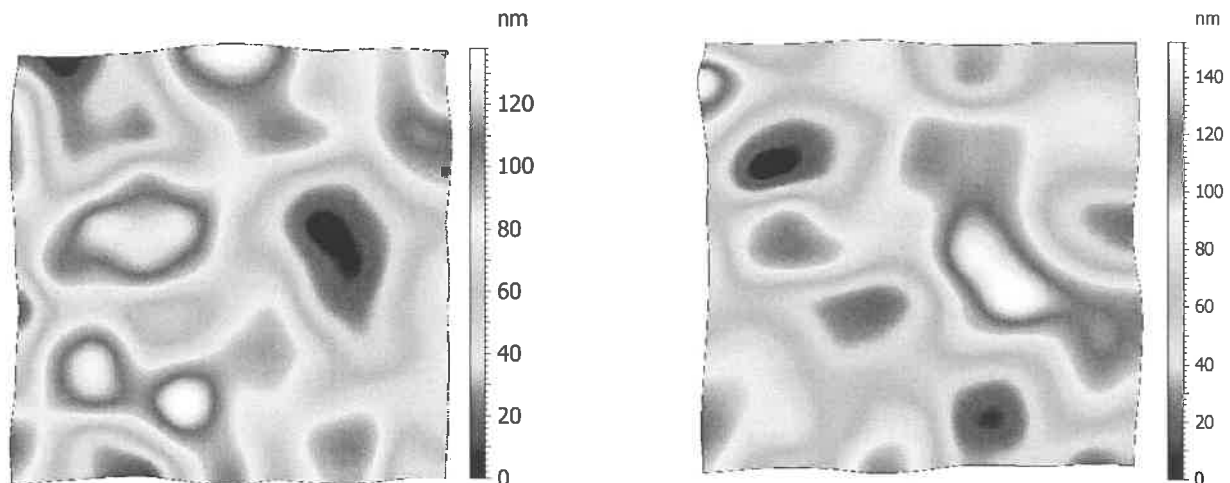


FIGURE 6. Surface data collected from the gauging face of a grade K gauge block at two different positions. The sides of the surfaces are 5 mm. Please note that the colour palette for this figure is only correct in the electronic version of this paper.

signals. This resulted in a calibration uncertainty due to sensor noise of up to 25 nm ($k = 1$) in certain situations. Further investigation into suitable low-noise band-pass filters, including amplification, resulted in a calibration uncertainty due to sensor noise of 8 nm ($k = 1$). Therefore, further development into suitable signal processing electronics and electrical connections has begun.

Throughout the testing stages, both off and on the CMM, several environmental factors proved detrimental to the operation of the probe. The sensitivity of the piezoelectric sensors and actuators is affected by changes in the relative humidity of the surrounding. It is estimated that slow changes in relative humidity can affect the length measuring capability of the vibrating micro-probe by 5 nm per 1 %rh.

During the completion of the repeatability measurement, there was a change in the environmental humidity of 2.4 %rh over 2.5 hours, which could result in a 12 nm change in the length measuring capability of the probe. During both flatness measurement tests, a change of approximately 1 %rh was observed over twenty-five minutes, which could result in a 5 nm change in the length measuring capability of the probe.

A further environmental factor that proved detrimental to the operation of the probe is the surrounding ambient light intensity. To mitigate this effect, the ambient light intensity was kept constant. Future enhancements to the design of the vibrating micro-probe should, therefore, include an opaque barrier layer on the piezoelectric elements to reduce moisture diffusion and to block any incident light.

The operational strategies of the probe should also be enhanced to account for small and gradual changes in environmental humidity and light intensity. This would involve automatically resetting the reference signal of the probe after every probing point is measured, or at regular intervals. This technique was applied manually during the measurement of the repeatability (four times in one thousand measurements) and flatness measurements (twice in one hundred measurements), suggesting that the humidity and light intensity changes would have a lesser effect on the results than previously described.

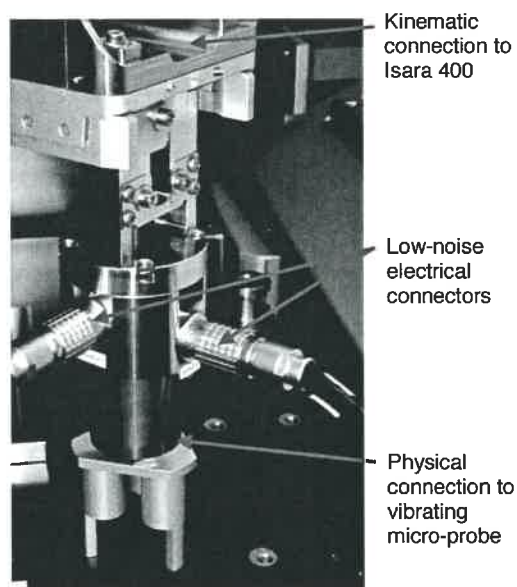


FIGURE 7. Newly designed, low mass, CMM head, incorporating low-noise connectors. The main bulk of the head is manufactured from Invar.

Certain geometrical limitations, due to the size and mass of the prototype CMM head (as shown in figure 4), were also apparent during testing. Therefore, a refined CMM head was designed and manufactured, suitable for further testing of the probe. The new CMM head is shown in figure 7.

CONCLUSION AND FUTURE WORK

A vibrating micro-CMM probe has been successfully interfaced with an ultra-precision CMM. The single-axis probing error of the resulting system has been determined, and a simple flatness measurement has been completed.

Future work in this area should initially focus on the verification of the 2D and 3D capability of the probe, according to previously defined procedures [7] [13] [14].

Also, throughout the testing stages, both off and on the CMM, it has become apparent that the ability of the vibrating micro-probe to counteract the surface interaction forces also enables it to detect, and possibly trigger from those same forces [1]. Therefore, the vibrating micro-probe should also be tested when operating in non-contact mode, to further investigate this mode of operation.

The ability of the vibrating micro-probe to operate in a non-contact mode will have far reaching implications, especially as the use of new materials become prevalent in precision manufacturing, for example organic materials, aerogels, novel coatings on optics or medical implants and structured surfaces on 3D objects.

ACKNOWLEDGEMENTS

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MACHINED WORKPIECE ERROR PREDICTION

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INTRODUCTION

The ability to make detailed, accurate predictions of actual workpiece errors is economically significant in multiple ways, affecting manufacturing costs (planning & set-up time, machine capability and maintenance), quality and time to market. Prediction of workpiece accuracy would allow design and manufacturing engineers to:

- Trade off feature tolerance choices against existing manufacturing process capabilities during design
- Develop machine specifications to meet required capabilities for new designs
- Eliminate non-value-added operations with "first part right" manufacturing process development
- Reduce risk in time-sensitive development and supply chain management
- Diagnose root causes rapidly for manufacturing process and maintenance problems
- Perform contract machining supplier qualification based on virtual capability analysis
- More effectively apply comprehensive virtual manufacturing technology

To be effective such predictions need to be feature-specific and directly related to design tolerances specified by standards such as ASME Y14.5 [1] or ISO 1101 [2]. By this we mean that predictions must be made for individual tolerances (e.g., cylindricity or true position) of individual workpiece features (e.g., a particular bore or slot.)

In the 1970s and '80s, two technical threads began that bear directly on the problem of machined workpiece error modeling. The first was the development of error budgeting methods [3] for systematically making estimates of the net effect of all of the errors in a machine or process. The second key development was the kinematic error model. [4]

In parallel, great strides have been made in the characterization and standardization of the parameters contributing to the accuracy of machine tools, measuring machines and other manufacturing equipment over the last 35 years.[4][5][6]

Slightly later, software systems to simulate machining processes using "destructive" solid geometry techniques began to be developed, and now have become available.

Unfortunately, limitations in each of these technologies prevent their simple combination to produce realistic predictions of the detailed geometry and surface form errors of machined workpieces.

OVERVIEW OF PRESENT WORK

To advance the state of the art for improved error prediction, Independent Quality Labs (IQL) and its technology partners began a dual path research program with the support of the US Army ARDEC. We report here on the successful completion of the two tracks of this project.

The first track (DFM) of this project focused on developing an error-budget-based tolerance consumption software tool which is relatively easy to use across a range of users (design, manufacturing, maintenance, quality, procurement) but delivers feature-specific predictions of sufficient accuracy to identify potential and actual manufacturing problems correctly.

The second track (SIM) of the project focused on more precise machining simulation using a significantly enhanced kinematic model of machine error motion to correctly predict global geometry and local surface form errors for all machined features. These model predictions were then used with commercial simulation software to produce realistic virtual part geometry, which was then compared to

measurements of actual machined workpiece geometry.

LOCUS® DFM DEVELOPMENT

The broad development goals for the DFM tool included:

- Tolerance/datum specific results
- Workpiece orientation specific results
- Problem-oriented simplified output
- Limited but extensible information about machine errors
- Computational efficiency
- Reliance on standards

The development of Locus® DFM to meet these goals had three major sub-tasks: a feature-specific error budgeting engine, realistic generic grades of machine tool performance and a relatively simple user interface.

Feature-specific Error Budgeting Engine

IQL previously developed an error budgeting process for analyzing feature-specific tolerance consumption based on an extensive list of measured performance errors identified in machine tool standards [5][6]. This process combines a kinematic error model with non-kinematic errors in an rss summation. The specific nature of the tolerance (type, datum structure, sensitive direction) is taken into account to select and use only those errors that are significant for the feature of interest. [3]

This basic analytic process forms the core technology for a fully automated, callable DFM computation engine in this sub-task. Key elements include the ability to use generic, estimated or measured machine errors, linearization of machine errors to reduce workpiece positioning sensitivity, and a methodology to correctly combine form and location effects in Attribute Feature Pairs (AFPs).

Generic Machine Performance Modeling

IQL has an extensive database of over 500 machine characterizations spanning many kinematic sub-classes and machine performance levels. For the DFM task, data mining in this database allowed development of realistic generic machine performance grades. This simplifies the need for machine performance knowledge by design users, while keeping the option of using all the mathematical

structure associated with individual machine errors, if they are actually known.

Five generic machine grades were identified in the data for two demonstration subclasses. Typical constants or error slopes (ppm errors) were assigned for the many individual errors. Those grades are summarized below.

TABLE 1. DFM's five generic grades

Performance Grade	Volume Diagonal Error Slope ($\mu\text{m/m}$)
AAA	<10
AA	10-20
A	20-40
B	40-65
C	>65

User Interface

The simplest way to capture the tolerancing details of workpiece features is to work from a CAD annotated solid model. A CAD environment is also typically the place where manufacturability analysis takes place. For these reasons, the Locus® DFM tool was initially implemented as an add-in to a commercial CAD system (SolidWorks®).

The DFM tool extracts all feature tolerances from the model and passes them sequentially to the DFM engine with a particular workpiece orientation on the machine. This process is repeated with all basic workpiece orientations plus specific orientations for hard-to-access features. The DFM engine returns predicted errors for each feature tolerance/orientation combination, and a tolerance consumption percentage is calculated.

The user is then presented with a rank ordered list of feature tolerances based on a user selected constraint (minimum orientation set or minimum error set). The ranking is based on the minimum machine grade required to keep the tolerance consumption below a selected percentage (e.g., 25%). This tolerance consumption limit is set by allocating in a global error budget what percentages of the feature tolerance may be consumed by the machine tool, process, inspection uncertainty and environmental influences.

DFM RESULTS

A typical output table is shown as a screen capture in FIGURE 1 below. Note that the first

item in the listing, "Diameter2" of "SimpleHole1", requires an AA grade machine to produce it at the 25% tolerance consumption level, while the "CompositePosition1" of the same feature only requires a B grade machine to hold the 25% tolerance consumption. All other features on this workpiece can be successfully produced on a C grade machine.

Best Orientation Minimum Orientations AFP Part Orientations Failed AFPs					
Attribute and features when using minimum orientations					
Attribute	Feature	Machine Grade	Orientation	Design Tolerance	Consp Level
Diameter2	Simple Hole1	AA	1	0.00050	25.0 %
CompositePosition1	Simple Hole1	B	3	0.00100	25.0 %
Diameter7	CounterSink3	C	3	0.00984	25.0 %
Depth5	CounterSink3	C	3	0.03937	1.0 %
Position6	CounterSink3	C	3	0.00500	1.2 %
Position5	CounterSink3	C	3	0.02000	4.0 %
AngleBetween5	Plane28	C	4	2.00000	0.4 %
DistanceBetween34	CounterSink3	C	3	0.03937	1.5 %
Diameter6	Simple Hole7	C	4	0.00250	17.6 %
DistanceBetween36	Plane38	C	6	0.02000	2.3 %
DistanceBetween35	Plane33	C	6	0.03937	1.2 %
Radius4	Cylinder3	C	1	0.01969	2.2 %
DistanceBetween1	Plane12	C	6	0.03937	1.0 %

FIGURE 1. Locus® DFM output in SolidWorks®

LOCUS® SIM DEVELOPMENT

Key developments required to produce a better simulation of machined accuracy are:

- Enhancement of standard kinematic models to incorporate significant non-kinematic effects seen in machine tools
- Methodology to handle tool orientation errors effects on form
- Technique to incorporate machine error effects into simulation software
- Validation of new approach by direct comparison of simulated virtual workpiece geometry with measurements of actual machined workpieces

Limitations of kinematic modeling

Kinematic error models make precise predictions of individual error effects on the trajectory of a tool point, but the collection of these trajectory points is not easily related to final part geometry for comparison to required tolerances. Nor does this modeling method include the effects of the multitude of non-kinematic errors, such as axis reversal errors, commonly seen in machine tools. Finally, this modeling technique does not completely handle the form errors that arise from misalignment of the tool (spindle) orientation with the part coordinate system's axes.

Enhanced Kinematic Modeling

Locus® SIM enhances a kinematic error model of the machine by directly incorporating several significant non-kinematic errors. The enhanced model modifies the dependence of angular errors on the location of a single axis to allow dependence on a cross-axis location. This handles non-kinematic effects of typically over-constrained machine tool guideways. Also, all kinematic errors are allowed to have motion-direction-dependent zero offsets to account for reversal errors. This enhanced model does not yet include spindle rotational motion errors, servo dynamics, table bending, tool deflection or thermal errors.

LOCUS® eM Machine Data Collection

Three specific enhancements of our machine error data collection process are important to note.

First, data was collected with a project sponsored system, Locus® eM, interfaced with a variety of measurement instruments. The system stored data files in formats consistent with the Draft B5.59 Information Technology for Machine Tools Parts 1 and 2 [7][8].

Second, since local bending of the workpiece mounting surface (i.e., the machine table) is not included in the current model, all machine error data are collected between the machine spindle and components mounted on a rigid sub-table (actually an aluminum spar) mounted quasi-kinematically on three points at the table guideway bearing locations.

Finally, we also address the often overlooked necessity to post-process the measurement data so the Abbe and Bryan errors have a common set of reference axes for internal consistency in the handling of axes' relative orientation errors, e.g., squarenesses.

Tool Orientation Errors

Proper handling of the tool orientation relative to the workpiece coordinate system is essential to model the workpiece form errors correctly. As stated, the standard kinematic model computes the location of a single tool point location. However, many tools (e.g., long end mills, shell mills, drills) cut simultaneously over a distributed region. Therefore, if the tool is slightly tilted relative to the part, it will make oval holes or inclined vertical faces. This effect is not usually represented by kinematic modeling but is an

important cause of workpiece error on most machine tools.

To handle this effect correctly, the extended kinematic model developed in this project also incorporates the extraction of the local tool orientation angles from the final composite error matrix for use in the model.

Software Integration

The sequence to make a virtual workpiece starts with a CAD nominal solid model generating a G-code program as post-processed output from a CAM system (MasterCAM®.) The enhanced kinematic model is incorporated into a post-post-processing module inserted between the CAM post-processor and a commercial machining simulation package (Vericut®.)

A critical detail of the post-post-processor is a step we call "gridding." The nominal G-code program contains the endpoints of machine nominal motions. However, the local form errors in machined features of a workpiece are caused by local motion errors between the start and endpoint. Therefore, nominal motion commands in the CAM produced G-code are subdivided into sequential short move segments. The error modeling computations are applied at each of these intermediate points to produce a realistic tool trajectory.

The post-post-processor output is thus a much larger G-code program with nominal motion replaced by error modified motion. The machining simulation software simply executes this modified G-code, but produces a simulated solid model representing what the machine tool can be expected to actually make. The errors in the simulated solid model were compared to the nominal model using measurement capabilities of the machining simulation software.

Validation

To validate this new simulation approach, a standard circle-diamond-square (CDS) was selected as a test workpiece. Using the original nominal G-code program for the CDS, three copies in aluminum were produced on our Haas VF-2 machining center. (The machine was intentional misadjusted to increase the magnitude of a few angular errors for testing purposes.) After scanning these test parts on a high accuracy CMM, the measured data were directly compared to surface points on the simulated solid model. Agreements between

simulated and measured results were within 0.010 mm, and generally smaller. All simulated geometry errors in the CDS are shown in FIGURE 2 with error magnitudes shown in pseudo-color.

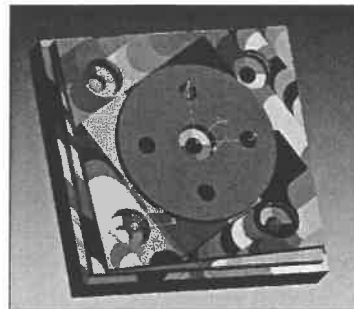


FIGURE 2. Simulated surface errors in CDS.

CONCLUSIONS

Locus® DFM and Locus® SIM proof-of-concept implementations demonstrate significant new capabilities for routine analysis of workpiece manufacturability at design time and during manufacturing process development and execution, as well as a significant improvement in the accuracy of virtual machining, showing realistic details of common form errors as well as dimensional geometry errors.

ACKNOWLEDGEMENTS

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FRICTION PARAMETER IDENTIFICATION AND COMPENSATION USING THE ELASTOPLASTIC FRICTION MODEL

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INTRODUCTION

Friction is one of the most important limiting factors for positioning accuracy in servosystems. In measurement applications, movements with velocities smaller than 1 mm/s are typical, acceleration values are equally small. Thus, friction force is the dominating force in these positioning mechanisms and has to be addressed by the control systems.

EXPERIMENTAL SETUP

The setup depicted in Figure 1, based on a linear motor, is used to position small objects like microlens-arrays or wafers containing MEMS relatively to optical sensors for the purpose of quality inspection. It is designed to be integrated into a modified Mahr MFU 100 measuring machine as an additional drive axis¹. The recirculating ball bearing guide induces substantial friction into the system. A linear encoder with 10 nm resolution is used to measure the position of the linear motor.

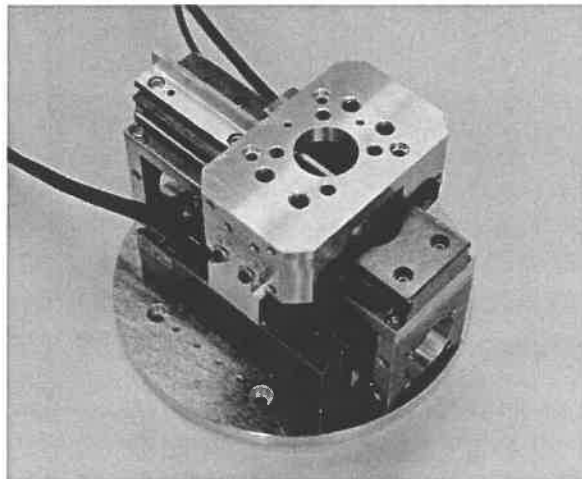


FIGURE 1. Linear drive: linear motor, recirculating ball bearing guide and incremental encoder.

¹Details are available on <http://www.isys.uni-stuttgart.de/forschung/mechatronik/nano/mt/index.html>

FRICTION MODEL

The elastoplastic friction model proposed in [1] was used to model and compensate for the friction of the measuring machine's drives. It is based on the bristle model of surfaces irregularities shown in Fig. 2.

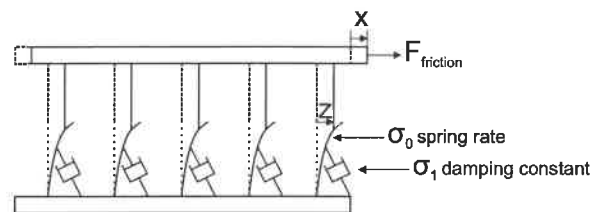


FIGURE 2. Bristle model of surface irregularities.

The surface irregularities causing friction are modeled as bristles with spring and damper characteristic as described in (1).

$$\dot{z} = \dot{x} \left(1 - \alpha(z, \dot{x}) \frac{z}{z_{ss}(\dot{x})} \right) \quad (1)$$

$$F_{\text{friction}} = \sigma_0 z + \sigma_1 \dot{z} + \sigma_2 \dot{x}, \quad \sigma_0, \sigma_1, \sigma_2 > 0$$

The spring rate is denoted by σ_0 , the damping constant by σ_1 , the viscous friction typically appearing at large velocities is described by σ_2 . The average bristle deflection z is the state variable of this first-order model. In contrast to other friction models, the elastoplastic model permits the modelling of motions in the pre-sliding domain as purely elastic motions without permanent displacement.

The friction force at small displacements, in the purely elastic or pre-sliding domain, is caused by the reaction force of the bristle deflection z . In this domain, the overall motion of the moving mass x is identical to the bristle deflection, $x = z$. When the motor force is reduced to zero, this elastic displacement will also disappear. As soon as the maximum purely elastic bristle deflec-

$$\alpha(z, \dot{x}) = \begin{cases} 0 & \text{for } |z| \leq z_{ba} \wedge \text{sgn}(\dot{x}) = \text{sgn}(z), \text{ pre-sliding domain} \\ \alpha_m(z, z_{ss}(\dot{x})) = \frac{1}{2} \sin\left(\pi \frac{z - \frac{1}{2}(z_{ss}(\dot{x}) + z_{ba})}{z_{ss}(\dot{x}) - z_{ba}}\right) + \frac{1}{2} & \text{for } z_{ba} < |z| < z_{ss}(\dot{x}) \wedge \text{sgn}(\dot{x}) = \text{sgn}(z), \text{ transition} \\ 1 & \text{for } |z| = z_{ss}(\dot{x}) \wedge \text{sgn}(\dot{x}) = \text{sgn}(z), \text{ sliding domain} \\ 1 & \text{for } |z| > z_{ss}(\dot{x}) \wedge \text{sgn}(\dot{x}) = \text{sgn}(z), \text{ Stribeck-Effect} \\ 0 & \text{for } \text{sgn}(\dot{x}) \neq \text{sgn}(z), \text{ motion reversal, purely elastic} \end{cases} \quad (2)$$

tion, the break-away deflection z_{ba} , is exceeded, sliding between the bristles occurs and the sliding domain is reached. The sliding motion w is a plastic displacement which will not disappear when the motor force is deactivated. The transition from the pre-sliding to the sliding domain is parametrized by the function $\alpha(z, \dot{x})$. As proposed in [2] and [1], $\alpha(z, \dot{x})$ is parametrized by (2). The bristle deflection increases in the sliding domain with increasing travel distance. It is asymptotically limited by the velocity-dependent maximum steady-state bristle deflection $z_{ss}(\dot{x})$ with $0 < z_{ba} \leq z_{ss}(\dot{x})$, $\forall \dot{x} \in \mathbb{R}$.

The velocity-dependence of $z_{ss}(\dot{x})$ describes the velocity-dependence of the friction force F_{friction} according to the *Stribeck-Curve* [3]. The *Stribeck-Curve* can be parametrized by (3) as described in [2].

$$z_{ss}(\dot{x}) = \frac{\text{sgn}(\dot{x})}{\sigma_0} \left(F_C + (F_S - F_C) e^{-\left(\frac{\dot{x}}{v_s}\right)^2} \right) \quad (3)$$

A typical *Stribeck-Curve* is depicted in Fig. 3. F_C denotes the *Coulomb-Friction* and F_S denotes stiction. The *Stribeck-Velocity* v_s is a form factor of the curve. Note that viscous friction is not included in the chosen representation of the *Stribeck-Curve* because it is already included in (1) by the term $\sigma_2 \dot{x}$.

FRICTION PARAMETER IDENTIFICATION

The microscopic friction parameters, the bristle parameters σ_0 , σ_1 , z_{ba} and $z_{ss}(\dot{x} = 0)$, are identified using trapezoidal motor force trajectories. The *Stribeck-diagram* is recorded using velocity trajectories. When the viscous friction coefficient σ_2 is determined, the remaining portion of the *Stribeck-diagram* (as shown in Fig. 3) can be fitted to the parametrization (3) with the parameters F_C , F_S and v_s . A detailed description is given in the following.

for $|z| \leq z_{ba} \wedge \text{sgn}(\dot{x}) = \text{sgn}(z)$, pre-sliding domain
for $z_{ba} < |z| < z_{ss}(\dot{x}) \wedge \text{sgn}(\dot{x}) = \text{sgn}(z)$, transition
for $|z| = z_{ss}(\dot{x}) \wedge \text{sgn}(\dot{x}) = \text{sgn}(z)$, sliding domain
for $|z| > z_{ss}(\dot{x}) \wedge \text{sgn}(\dot{x}) = \text{sgn}(z)$, *Stribeck-Effect*
for $\text{sgn}(\dot{x}) \neq \text{sgn}(z)$, motion reversal, purely elastic
(2)

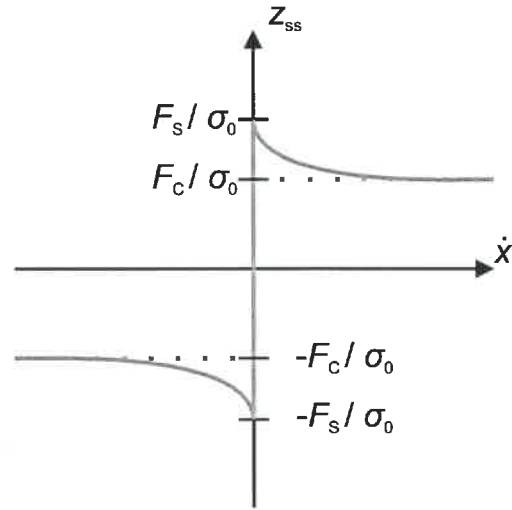


FIGURE 3. *Stribeck-Curve*.

Stribeck-diagram

Assuming that the *Stribeck-curve* does not vary significantly over the length of the linear guide, an interval $x_i = \pm 5 \text{ mm}$ around the position $x = 0$ was chosen to identify the macroscopic friction parameters. The motor is operated in closed loop position control and driven through this interval at constant speed, since the *Stribeck-curve* describes the steady-state friction force as a function of velocity. The position reference trajectory is depicted in Figure 4.

The reversal points are placed further away from x_i for increasing velocities, in order to ensure that all acceleration transients have decayed when the motor enters x_i . Thus, the control effort inside x_i is only required to overcome the friction. (4) can be used to calculate the actual friction force F_{friction} from the control effort (motor force) F_{motor} .

$$F_{\text{friction}} = F_{\text{motor}} - m\ddot{x} \quad (4)$$

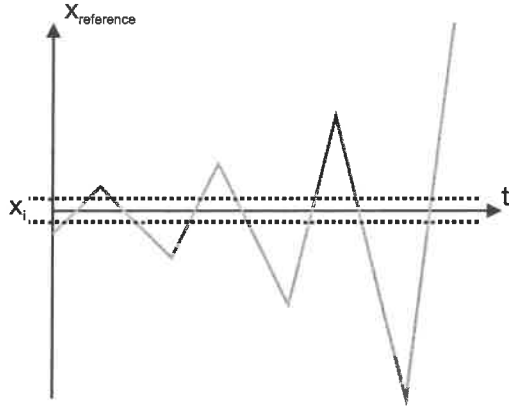


FIGURE 4. Position reference trajectory used to measure the Stribeck-Curve.

The term $m\ddot{x}$ (including the motor inertia m) is included to correct for any velocity deviations, although experience shows that their contribution to the motor force can be neglected.

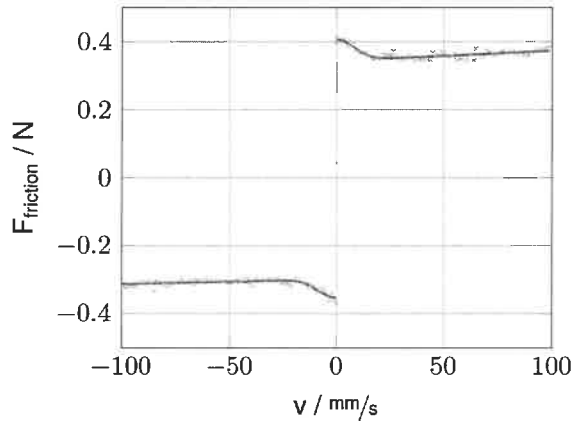


FIGURE 5. Stribeck-Curve of the linear drive.

Each time the motor passes through x_i , one sample of the *Stribeck*-curve is taken by calculating (4) and averaging the determined friction force over x_i . The data points for velocities from 0.01 mm/s to 100 mm/s, plotted over the positive (or negative) velocity of the motor, result in the *Stribeck*-curve depicted in Figure 5. The parameters σ_2 , F_C , F_S and v_S of the parametrization (3) can be determined by fitting this curve to the samples using least-squares. The parametrization is also included in Figure 5 and the parameters can be found in Table 1.

TABLE 1. Parameters of the Stribeck-curve (3).

$\dot{x} < 0$		$\dot{x} > 0$	
parameter	value	parameter	value
F_C	-0.298 N	F_C	0.343 N
F_S	-0.352 N	F_S	0.406 N
σ_2	0.141 $\frac{\text{N}}{\text{m/s}}$	σ_2	0.296 $\frac{\text{N}}{\text{m/s}}$
v_S	-0.011 m/s	v_S	0.012 m/s

Parameters of the Bristle Model

The bristle parameters, σ_0 , σ_1 , z_{ba} and $z_{ss}(\dot{x} = 0)$, are identified by applying the force trajectories with increasing maximum values shown in Figure 6 to the motor in open loop operation. The measured motor position x is again composed of the elastic portion z which is equivalent to the bristle deflection and the plastic portion w , $x = z + w$. Since there is no measurement for z available, it has to be reconstructed based on the following consideration:

If the force slope is chosen small, the portion of the motor force caused by the motor's inertia can be neglected, thus the motor force is approximately equal to the friction force, $F_{\text{friction}} \approx F_{\text{motor}}$. Furthermore, the microscopic and the macroscopic damping, represented by σ_1 and σ_2 , can be neglected. The spring force of the bristles, the *Stribeck*-effect and the slip (for $z > z_{ba}$) remain. As can be seen in Figure 6, the elastic displacement z can be calculated after each motion cycle as the difference of maximum displacement x_{max} and the plastic displacement $x_{\text{end}} = w$, $z = x_{\text{max}} - w$. The bristle parameters can thus be identified from the relaxation of the elastic displacement at the end of each motion cycle.

As long as the motion is within the purely elastic regime $x_{\text{max}} \leq z_{ba}$, there will be no displacement after the end of the motion cycle, $w=0$. The diagram in Figure 7 shows z_{max} plotted against x_{max} . For small displacements, the diagram is a straight line since $z_{\text{max}} = x_{\text{max}}$. The end of this straight line indicates the end of the purely elastic regime z_{ba} , which can thus be identified from this plot.

For larger motion amplitudes $x_{\text{max}} \gg z$, the elastic displacement z converges against the maximum bristle deformation z_{ss} , which can thus also be identified from Figure 8.

Figure 9 shows z_{max} plotted against F_{max} for the purely elastic regime. The linear spring-like behaviour of the bristles is obvious and the

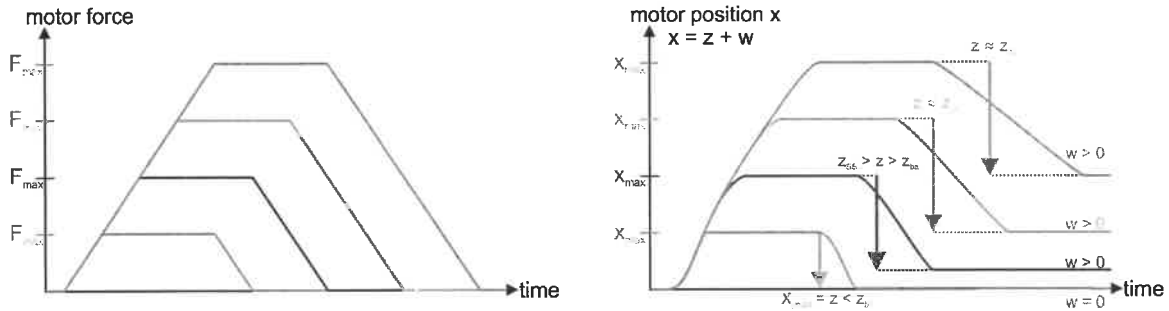


FIGURE 6. Force trajectories used to identify the bristle parameters.

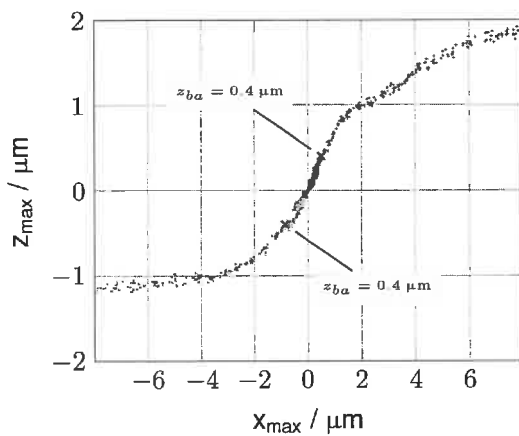


FIGURE 7. $z_{\max}$ plotted against $x_{\max}$ with z_{ba} indicated.

spring rate σ_0 can be derived from this plot as $\sigma_0 = \Delta F / \Delta z$.

In a final step, the bristle damping constant σ_1 can be derived from considerations on the overall system damping ratio ζ , (5).

$$\zeta = \frac{d}{2\sqrt{km}} = \frac{\sigma_1 + \sigma_2}{2\sqrt{km}} \quad (5)$$

Since mechanical systems with a large influence of friction exhibit a highly damped system response, it is a good assumption that the damping ratio ζ is large, so $\zeta = 1$ is chosen. σ_1 is thus calculated from (6).

$$\sigma_1 = 2\zeta\sqrt{\sigma_0 m} - \sigma_2 \quad (6)$$

An overview of the identified system parameters is given in Table 2. Direction-dependent param-

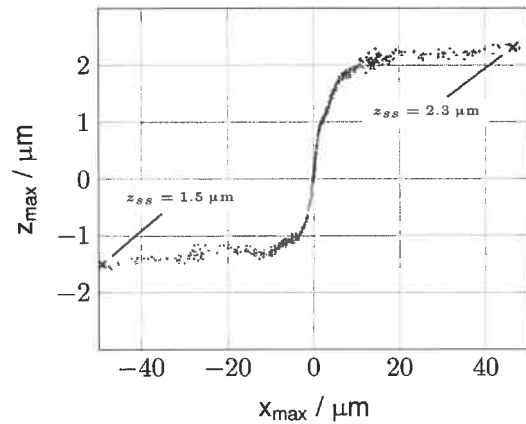


FIGURE 8. $z_{\max}$ plotted against $x_{\max}$ with z_{ss} indicated.

eters are represented by their mean values in this table.

TABLE 2. Dynamic parameters of the linear drive.

parameter	value
z_{ba}	0.4 μm
z_{ss}	1.927 μm
σ_0	65922 N/m
σ_1	479.36 $\frac{\text{N}}{\text{m/s}}$
m	0.843 kg

FRICITION COMPENSATION

The goal was to implement a position control system allowing for high-accuracy movements in both the pre-sliding and the sliding domain with one single controller, avoiding switching between separate controllers for the two domains.

To achieve this goal, the elastoplastic friction model is used in a feedforward control scheme, in which the estimated friction force $\hat{F}_{\text{friction}}$ is ap-

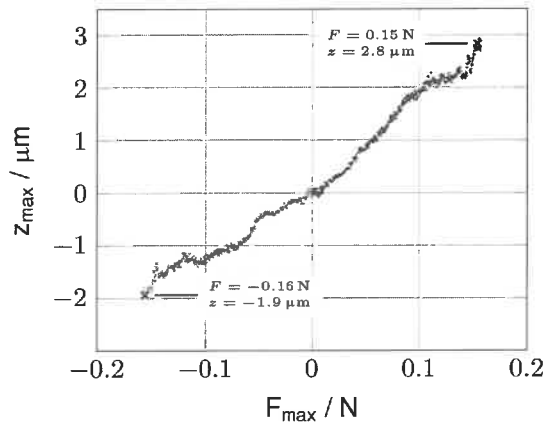


FIGURE 9. z_{max} plotted against F_{max} .

plied to the motor in order to compensate for the friction [2]. An estimator for the unknown initial state of the friction model, the initial bristle deflection z , is established by using the feedback controller's control effort, which is equivalent to the difference between the estimated motor force and the actual motor force as feedback for the estimator loop. Since transitions between the pre-sliding and the sliding domain are included in the elastoplastic friction model, they are also included in the estimation and feedforward compensation. In addition to the mentioned observer for microscopic motions, a second observer for macroscopic motions is included into the friction estimator to adapt the *Stribeck*-diagram to any position-dependent variations along the linear bearing and wear. The feedback control effort is also used as feedback for this estimator, however the estimation is only active at large velocities.

Assuming that friction forces are sufficiently compensated for by this feedforward controller, the feedback controller can simply be designed for the remaining linear "free mass" plant. However, the linear portion of the friction force, the viscous friction $\sigma_2 \dot{x}$, is also included in the linear plant to obtain a more accurate design model for the feedback controller.

A PID feedback controller, combined with a linear flatness-based feedforward controller for improved trajectory tracking, is used for this purpose. The control effort for the linear plant is thus calculated from (7). The overall control system is shown in Figure 10.

$$u = k_I \int e \, dt + k_P e + k_D \dot{e} + \underbrace{\sigma_2 \dot{r} + m \ddot{r}}_{\text{linear feedforward}} \quad (7)$$

EXPERIMENTAL RESULTS

The motion of the drive axis at a slow measuring speed of $200 \, \mu\text{m/s}$ is regarded to demonstrate the effectiveness of friction compensation and motion control. The position trajectory, generated with the point-to-point trajectory generator, and the measured position is shown in Figure 11. The control error during this motion with a maximum value of $0.5 \, \mu\text{m}$ is depicted in Fig. 12. The control error of more than $2 \, \mu\text{m}$ without feedforward friction compensation depicted in Figure 13 is four times as large.

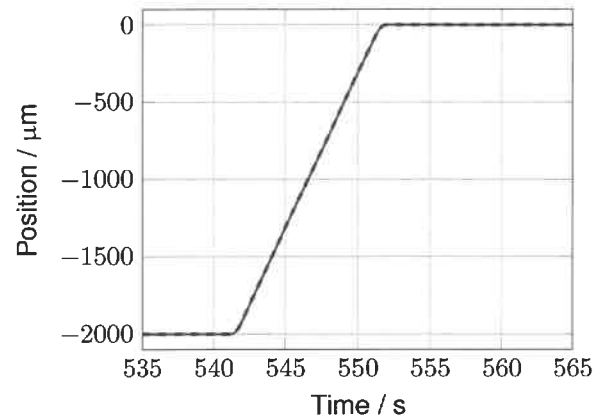


FIGURE 11. Reference trajectory and measured position with feedforward friction compensation.

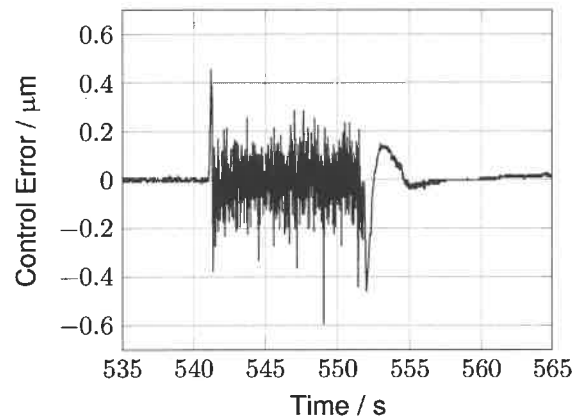


FIGURE 12. Position control error with feedforward friction compensation.

Fig. 14 depicts the control effort during the motion. It is obvious that the overall control effort

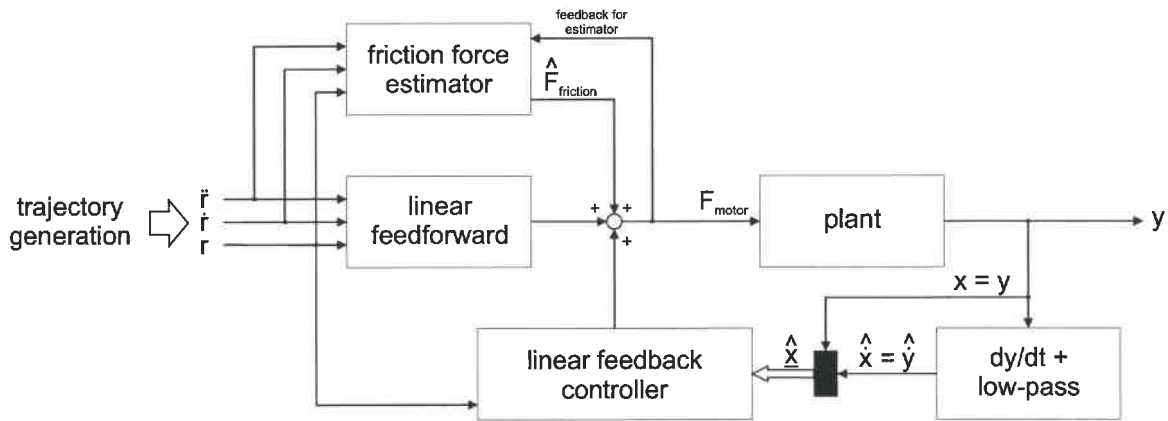


FIGURE 10. Overview of the control system architecture.

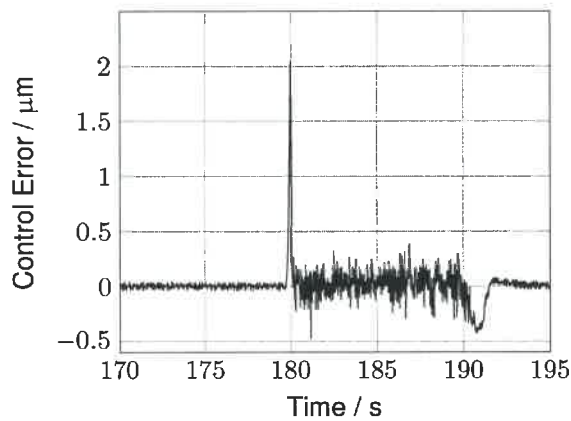


FIGURE 13. Position control error without feedforward friction compensation.

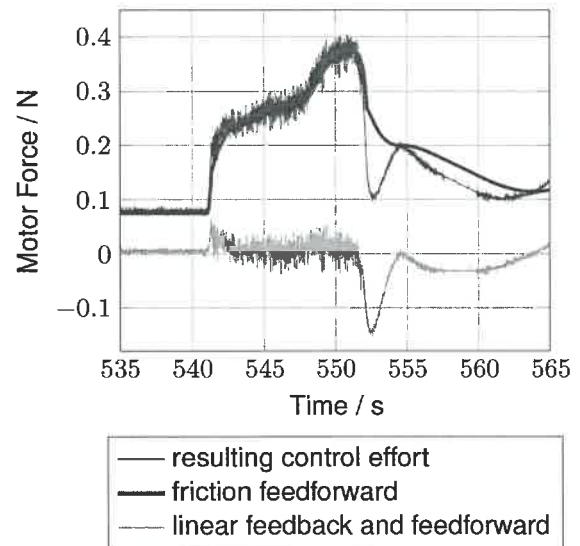


FIGURE 14. Control effort with feedforward friction compensation.

is dominated by the friction feedforward compensator and that the feedback controller only has to compensate for small model uncertainties and disturbances. This shows that the friction in the bearings and guides is estimated and thus compensated for well by the friction estimator. It also emphasizes that friction is the dominating force in motions with low speed as required to scan a surface with optical sensors, for example.

ACKNOWLEDGEMENTS

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DATA-DRIVEN CONTROL STRATEGY FOR A RETICLE STAGE IN A LITHOGRAPHIC TOOL

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INTRODUCTION

High-performance motion systems, such as lithographic wafer-scanners, usually require ultra-precision multi degrees-of-freedom (DOFs) motion control. In contrast to model-based control, data-driven control is widely used to address the practical difficulty on obtaining accurate system models. The rationale of data-driven control is that the controller is parameterized and the parameter vector is directly generated by fully using of the IO collection data, without the need of dynamics modeling. Most data-driven methods, such as IFT [1], CbT [2] and FDT [3], solve the controller design problem by iteratively minimizing an H_2 performance criterion, and the iterative law can be expressed as [1]

$$p^{k+1} = p^k - \gamma^k R^{-1} \nabla J|_{p^k} \quad (1)$$

where p is the controller parameter, ∇J is the gradient of the performance criterion J with regard to the parameter vector p , k is the number of iteration, R is a positive definite matrix and γ is the step size.

In this paper, a simplicity and accuracy oriented data-driven control strategy is proposed to synthetically build the control system of a 6-DOF ultra-precision dual-stroke stage. The stage, namely reticle stage which is used in a lithographic tool to manufacture integrated circuits (ICs), moves the pattern at nanometer accuracy in synchronization with the wafer [4].

DESIGN AND CONTROL CONCEPT

The developed dual-stroke reticle stage, consisting of a 6-DOF short-stroke (SS) stage for fine motion and two 1-DOF long-stroke (LS) stage for coarse motion, is used to achieve high-velocity and high-precision motion performance.

Figure 1 shows the structural configuration of the dual-stroke stage. SS stage is driven by ten voice coil motors (VCMs) to achieve contactless 6-DOF movement. Considering the heat dissipation, four magnetic gravity compensators (MGCs) are mounted in the center of the four vertical VCMs to counteract the gravity. The displace-

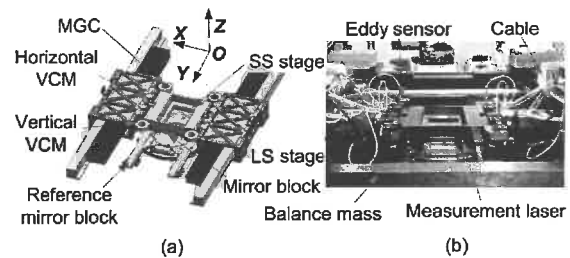


FIGURE 1. (a) The schematic and; (b) The real assembly of the dual-stroke reticle stage.

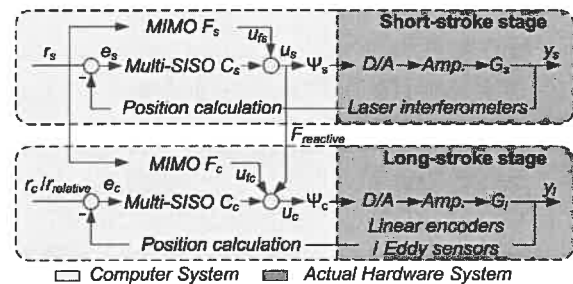


FIGURE 2. Control scheme of the system

ment is measured by multi-axis laser interferometers with 0.6 nm resolution.

Two friction-free LS stages are symmetrically distributed on the two sides of SS stage. Each LS stage is driven by an ironless linear motor and the measurement system includes eddy sensors and linear encoders.

Figure 2 shows the concept of the control system. SS stage is measured by laser interferometers and is controlled in the absolute coordinate system. The LS loop is synchronized with SS stage, which has two control modes: one is to keep a constant relative displacement with SS stage, and the other is to track the same trajectory of SS stage. LS stage is measured by eddy sensors under the first mode, and is measured by linear encoders under the second mode.

The underlying performance with the used control configuration is mainly determined by three parts: 1) the static input dynamic decoupling matrix Ψ for achieving closed-loop diagonal dominance; 2) multi-SISO feedback controller C for

fine robust stability and quick response; 3) MIMO feedforward controller F for improving tracking performance.

THE SYNTHETICAL DATA-DRIVEN ITERATIVE CONTROL STRATEGY

Figure 3 shows the horizontal dynamics of SS stage, and the other three DOFs have the similar dynamic characteristics. As the dynamics present the high-order and uncertainty characteristics which are adverse to employ model-based approach, a data-driven control strategy is proposed with an integrated design for Ψ_s , C_s and F_s . For simplicity, SS stage is considered in this paper.

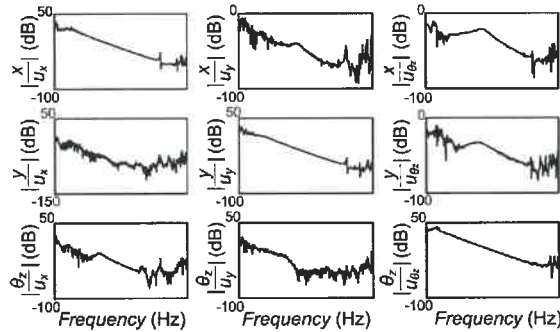


FIGURE 3. Horizontal dynamics of SS stage

In this paper, considering the mismatch between the achieved and the desired output is the crucial performance index, the H_2 performance criterion is chosen as

$$J(p) = \frac{1}{N} \sum_{n=0}^{N-1} E[\tilde{y}(n, p)^2] = \frac{1}{N} \sum_{n=0}^{N-1} E[(y - y_d)^2] \quad (2)$$

where y and y_d denote the achieved and the desired output, respectively. The gradient can be derived as

$$\nabla J(p) = \frac{2}{N} \sum_{n=0}^{N-1} [\tilde{y}(n, p) \nabla y(n, p)] \quad (3)$$

Generally, R in (1) is chosen as the Hessian matrix and the Gauss-Newton direction [5] is a common choice as follow

$$R(p) = \frac{2}{N} \sum_{n=0}^{N-1} [\nabla y(n, p) \nabla y(n, p)] \quad (4)$$

As $y(n)$ can be directly collected from the system, the approximation of ∇y is crucial for the control performance. The following subsections will present the proposed design method of the Ψ_s , C_s and F_s , and the corresponding approximation principle of the gradient, respectively.

Static Input Decoupling Matrix Ψ_s

Consider the plant with linear assumption

$$M\ddot{X} + C\dot{X} + KX = F_s \quad (5)$$

where M , C and K denote the inertial, damping and stiffness matrix, respectively. The state vector X is defined as $[x, y, z, \theta_x, \theta_y, \theta_z]^T$, and the global force F_s is defined as $[F_{x1}, F_{y1}, F_{z1}, T_{x1}, T_{y1}, T_{z1}]^T$. Using the Moore-Penrose inverse that obtains the minimum power commutation, the global force F_s , the local force f and the control force u_s satisfy

$$F_s = (\Psi_s^*)^+ f_s = (\Psi_s^*)^+ \Psi_s u_s \quad (6)$$

where f_s is $[f_{x1}, f_{x2}, f_{y1}, f_{y2}, f_{y3}, f_{y4}, f_{z1}, f_{z2}, f_{z3}, f_{z4}]^T$, u_s is $[u_x, u_y, u_z, u_{\theta_x}, u_{\theta_y}, u_{\theta_z}]^T$ and Ψ_s^* is the optimal value of the static input decoupling matrix Ψ_s . Ψ_s can be defined as

$$\Psi_s = \begin{bmatrix} 1 & 1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 1 & 1 & 1 & 1 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 1 & 1 & 1 & 1 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ -l_{x1y} & -l_{x2y} & -l_{y1x} & -l_{y2x} & -l_{y3x} & -l_{y4x} & -l_{z1x} & -l_{z2x} & -l_{z3x} & -l_{z4x} \\ l_{x1z} & l_{x2z} & -l_{y1z} & -l_{y2z} & -l_{y3z} & -l_{y4z} & 0 & 0 & 0 & 0 \end{bmatrix}^+ \quad (7)$$

where l_{ij} denotes the arm of force. In general, Ψ_s is determined by the computer-aided model in practical system. However, due to the inevitable modeling error, the offset $p_{\Psi} = [p_x, p_y, p_z]$ between the actual and theoretical center of mass (COM) will render additional torque $T^{ad} = [T_x^{ad}, T_y^{ad}, T_z^{ad}]$, as shown in Figure 4. To achieve diagonal dominance, a data-driven method is proposed to reduce the coupling. As the servo error e_s directly reflects the torque coupling, the desired output y_d in (2) is chosen as the reference profile r_s . With the rotational DOFs keep positioning, the rotation error can be approximated as

$$\begin{bmatrix} e_{s\theta_x} \\ e_{s\theta_y} \\ e_{s\theta_z} \end{bmatrix} = -\text{diag} \left(\frac{P_{\theta_x}}{1 + P_{\theta_x} C_{\theta_x}}, \frac{P_{\theta_y}}{1 + P_{\theta_y} C_{\theta_y}}, \frac{P_{\theta_z}}{1 + P_{\theta_z} C_{\theta_z}} \right) \begin{bmatrix} T_x^{ad} \\ T_y^{ad} \\ T_z^{ad} \end{bmatrix} \quad (8)$$

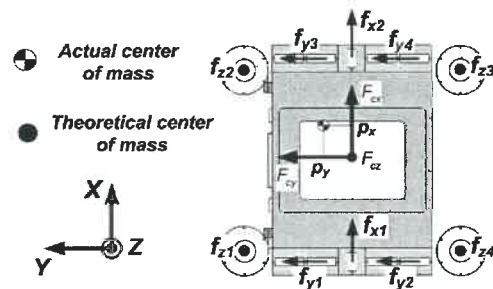


FIGURE 4. Force acted on the SS stage

where $T_{\Psi p}$ is the process sensitivity transfer function matrix. The additional torque matrix is

$$\begin{bmatrix} T_x^{ad} \\ T_y^{ad} \\ T_z^{ad} \end{bmatrix} = \underbrace{\begin{bmatrix} 0 & -u_z & u_y \\ u_z & 0 & -u_x \\ -u_y & u_x & 0 \end{bmatrix}}_{\Theta_{\Psi u}} \underbrace{\begin{bmatrix} p_x \\ p_y \\ p_z \end{bmatrix}}_{p_{\Psi}} \quad (9)$$

The gradient of the error e_s can be deduced as

$$\nabla e_s \approx -T_{\Psi p} \Theta_{\Psi u} \quad (10)$$

The force matrix $\Theta_{\Psi u}$ can be directly acquired from system data. Nevertheless, as the transfer function $T_{\Psi p}$ includes the information of the plant model, it cannot be directly obtained from system data. In this paper, a data-driven approximation based on the Toeplitz matrix is proposed to estimate $T_{\Psi p}$, and the principle of the Toeplitz matrix is presented in the appendix. Considering the existing high-order dynamics and inherent model uncertainty, $T_{\Psi p}$ is obtained by the data-driven approach. Specifically, as the Toeplitz matrix $T_{\Psi p}(ii)$ is the map between the i -direction process input u_{Ψ} and the i -direction output y_s , the Markov parameters of $T_{\Psi p}(ii)$ can be acquired by recording the corresponding output time-series of the 3-DOF ($\theta_x, \theta_y, \theta_z$) when a unit impulse is injected at the process input point. With the force matrix $\Theta_{\Psi u}$ and the Toeplitz matrix $T_{\Psi p}$, the gradient ∇e_s can be approximated. Therefore, combined with the Gauss-Newton direction, the iteration will conduct the theoretical COM converging to the actual COM.

Multi-SISO Feedback Controller C_s

In this subsection, the proposed tuning strategy of the feedback controller C_s is presented. With diagonal dominance achieved by Ψ_s , the system of each DOF can be simply expressed as

$$y_s(n) = P_s(z)u_s(n) + v_s(n) \quad (11)$$

where $u_s(n)$, $y_s(n)$ and $v_s(n)$ are the input signal, the output signal and the exogenous quasi-stationary disturbance, respectively.

In this paper, C_s is assumed to be linear and is defined as

$$C_s(z, p_c) = p_c^T \Gamma_c(z) \quad (12)$$

where p_c is the parameter vector and $\Gamma_c(z)$ is the fixed basis function of $C_s(z, p_c)$.

Generally, most practical feedback controllers adopt the linear structure, such as PID. Besides,

with Taylor expansion theory, any stable controller can be approximated by the linear structure. Therefore, without loss of generality, we employ the linear structure as C_s .

Thus, the mismatch between the achieved and the desired output can be deduced as

$$\begin{aligned} \tilde{y}(p_c) &= y - y_d = \frac{PC_s}{1+PC_s}r - \frac{PC_{sd}}{1+PC_{sd}}r \\ &= T_c(p_c)r - T_{cd}r \end{aligned} \quad (13)$$

where C_{sd} denotes the ideal controller that minimizes $J_c(p_c)$. Thus, the gradient of the output $\nabla y(p_c)$ is derived as

$$\nabla y(p_c) = \frac{P}{1+PC_s} \frac{1}{1+PC_s} \Gamma_c^T r = T_{cp} S_c \Gamma_c^T r \quad (14)$$

In this paper, we assume C_{sd} also possesses the same linear structure of C_s in (12), which guarantees the existence of the optimal parameter vector p_{cd} . Therefore, the gradient of the performance criterion can be expressed as

$$\begin{aligned} \nabla J_c &= \frac{2}{N} (\nabla y)^T \tilde{y} = \frac{2}{N} (T_{cp} S_c \Gamma_c^T r)^T T_{cp} S_{cd} \Gamma_c^T r (p_c - p_{cd}) \\ &= M_c (p_c - p_{cd}) \end{aligned} \quad (15)$$

where the optimal sensitivity $S_{cd} = 1/(1+PC_{sd})$.

Based on (15), we propose a novel iterative law which aims at directly achieving the desired performance with several iterations. The iterative law is designed as

$$p_c^{k+1} = p_c^k - \gamma^k M_c^{-1} \nabla J_c|_{p_c^k} \quad (16)$$

The iterative law has similar form with Gauss-Newton algorithm. However, Gauss-Newton algorithm, based on the second order Taylor expansion which ignores the high order terms, relies on the initial condition. Theoretically, the proposed iterative method can directly obtain the optimal parameter p_{cd} with one-shot execution, without the restriction of the initial condition.

Besides y_s , the iterative law (16) needs to obtain T_{cp} and S_c which determine M_c and ∇y_s . In this paper, we employ the Toeplitz matrix that has been introduced to approximate the variables.

MIMO Feedforward Controller F_s

To further improve the underlying tracking performance after tuning the feedback loop with Ψ_s and C_s , the feedforward controller F_s is of importance. In this paper, a data-driven decoupling feedforward control is proposed. The criterion is

chosen as the quadratic of the servo error e_s by choosing y_d as r_s in (2), since e_s is the evaluation index of the tracking performance. The feedforward controller is defined as

$$F_s = \begin{bmatrix} F_{ii} & F_{ij} \\ F_{ji} & F_{jj} \end{bmatrix}_{6 \times 6}, i, j \in \{x, y, \dots, \theta_z\}, i \neq j \quad (17)$$

To make the optimal problem convex, the controller structure consists of finite impulse response (FIR) filters, and is developed as

$$F_{ij}(p_{\bar{n}j}) = p_{\bar{n}j}^T \Gamma_f = p_0^j + p_1^j z^{-1} + \dots + p_r^j z^{-r} \quad (18)$$

There are several issues considered for choosing the controller. Firstly, FIR filter is a polynomial which is able to approximate the inverse model. Secondly, linear characteristic of the controller makes the overall servo error as an affine space with regard to the parameter vector p_f . Thirdly, the structure has a benefit to obtain the gradient. Once the term $\nabla e(p_0^j)$ is obtained, the whole gradient $\nabla e(p_{\bar{n}j})$ can be calculated immediately since the term $\nabla e(p_n^j)$ is equal to the term $\nabla e(p_0^j)$ by a delay of n sampling times, which $\nabla e(p_n^j) = \nabla e(p_0^j) z^{-n}$. This feature simplifies the practical implementation of the algorithm.

For the 6-DOF reticle stage control system, the servo error e_s can be described as

$$e_s(p_f) = S_f(r - v) - T_{fp} u_{fs}(p_f) \quad (19)$$

where

$$S_f = \begin{bmatrix} S_{xx} & \dots & S_{x\theta_z} \\ \vdots & \ddots & \vdots \\ S_{\theta_z x} & \dots & S_{\theta_z \theta_z} \end{bmatrix}, T_{fp} = \begin{bmatrix} T_{xx} & \dots & T_{x\theta_z} \\ \vdots & \ddots & \vdots \\ T_{\theta_z x} & \dots & T_{\theta_z \theta_z} \end{bmatrix} \quad (20)$$

and

$$u_{fs}(p_f) = \begin{bmatrix} F_{xx} & \dots & F_{x\theta_z} \\ \vdots & \ddots & \vdots \\ F_{\theta_z x} & \dots & F_{\theta_z \theta_z} \end{bmatrix} \begin{bmatrix} r_x \\ \vdots \\ r_{\theta_z} \end{bmatrix} \quad (21)$$

Meanwhile, the i -direction of $u_{fs}(p_f)$ can be further derived as

$$\begin{aligned} u_{fs}^i(p_{\bar{n}i}) &= \sum_{k=x}^{\theta_z} F_{ik}(p_{\bar{n}k}) r_k = \Pi_f p_{\bar{n}i} \\ &= \begin{bmatrix} r_x & \dots & r_x z^{-r} & \dots & r_{\theta_z} & \dots & r_{\theta_z} z^{-r} \end{bmatrix} \begin{bmatrix} p_{\bar{n}ix} \\ \vdots \\ p_{\bar{n}i\theta_z} \end{bmatrix} \end{aligned} \quad (22)$$

Thus, the gradient of the servo error e_s derived from equation (19) can be then expressed as

$$\nabla e_s(p_f) = -T_{fp} \hat{\Pi}_f \quad (23)$$

where

$$\hat{\Pi}_f = \begin{bmatrix} \Pi_f & & 0 \\ & \ddots & \\ 0 & & \Pi_f \end{bmatrix}_{6N \times 36(r+1)} \quad (24)$$

In this paper, the proposed control strategy employs the Toeplitz matrix to estimate T_{fp} , which is approximated by the data-driven approach. Besides, $\hat{\Pi}_f$ is only based on the reference profile and the structure of the feedforward controller. Thus, the gradient of the servo error $e_s(p_f)$ can be easily obtained.

EXPERIMENTAL ASSESSMENTS

In this section, the performance of the proposed data-driven iterative control strategy is evaluated on the developed reticle stage. The Versa Module Eurocard (VME) bus is used as a communication backbone among components. The VME PC (MPC7455) is used as the kernel to conduct the overall control system. The digital signal processor (DSP) (TMS320C6713B) acts as the task performer to process the digital data. The digital control system is harmonized with a sampling period of $T_s = 200$ us.

The validity of the data-driven strategy including the tuning of Ψ_s , C_s and F_s has been verified by three experiments, respectively. The three important parts of the applied motion control system are tuned synthetically to improve the underlying servo performance.

Optimization of Ψ_s

The proposed data-driven tuning strategy for Ψ_s is employed to attenuate additional torque that caused by the mismatch between the theoretical COM and the actual COM. To embody the coupling caused by the additional torque, the 1000nm step responses of the translational DOFs are conducted simultaneously. Figure 5 shows the performance of the proposed tuning method. It indicates that the coupling is obviously reduced after one-shot iteration, demonstrating its effectiveness. However, the coupling cannot be compensated completely due to the structure of Ψ_s is unable to deal the high-order dynamic characteristics.

The proposed method and the comparison [6] are conducted, as shown in Figure 6. Starting with the same initial condition, the proposed method is closer to the desired output after several iteration trials. Figure 7 presents the evolution of the value of $J_c(p_c)$ with the proposed method, the value of $J_c(p_c)$ with the proposed method reduces to 11, almost 35 times smaller than the initial value, indicating the favorable convergent performance of the proposed method.

$$C_s(p_c) = \begin{bmatrix} k_p & k_i & k_d \end{bmatrix} \begin{bmatrix} 1 & z^{-1} & z^{-2} \end{bmatrix} \quad (25)$$

In this paper, only the optimization performance of the x-direction is presented due to the space limitations. The x-direction of the stage is controlled by a standard PID controller as

Optimization of C_s

FIGURE 6. Performance of the proposed and comparison method with 200nm step response.

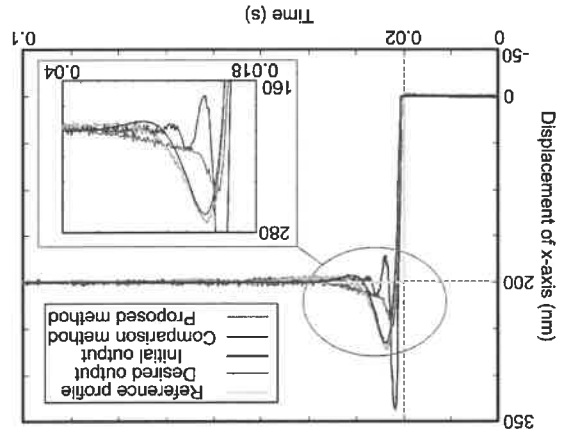
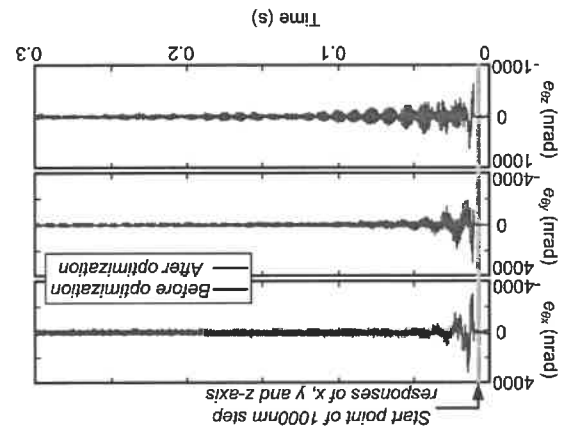


FIGURE 5. Responses of the rotation errors after the translational DOFs perform 1000nm step.



As y-direction of the stage is the working direction, y-DOF tracks the trajectory and other DOFs keep positioning in the experiment. Considering the number of the main resonance peaks, the order $n+1=6$ of the FIR feedforward controller offers a good compromise. Figure 8 shows the tracking error before and after optimization. It can be observed that the

Optimization of F_s

In addition, we think there have several reasons that the criterion $J_c(p_c)$ could not be further reduced: 1) the ubiquitous noise affects the estimation property in practical system; 2) the desired closed-loop model cannot be achieved by the PID controller structure.

FIGURE 8. Tracking error before and after optimization.

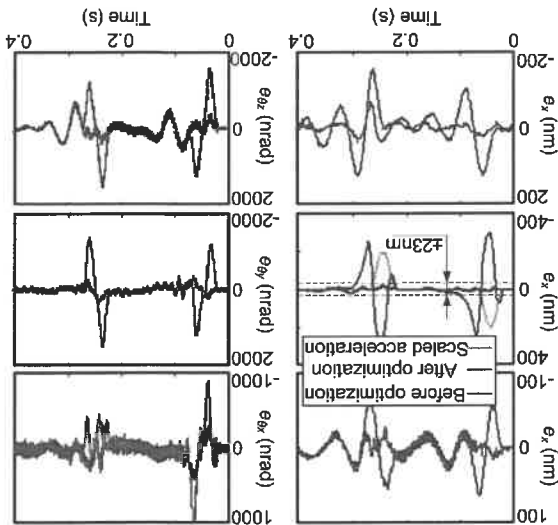
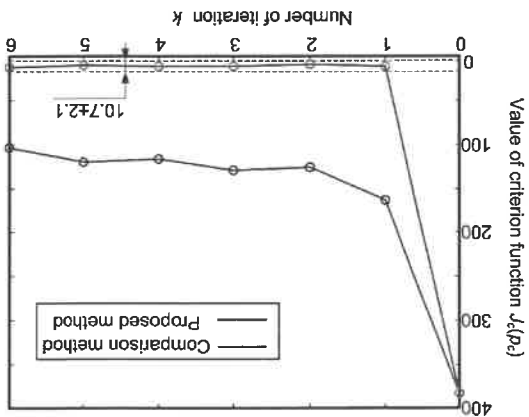


FIGURE 7. Value of the criterion $J_c(p_c)$ in the first six iteration trials



$$\begin{bmatrix} y(0) \\ y(1) \\ \vdots \\ y(n-1) \end{bmatrix} = \begin{bmatrix} D & 0 & \cdots & 0 \\ CB & D & \ddots & \vdots \\ \vdots & \vdots & \ddots & 0 \\ CA^{n-2}B & \cdots & CB & D \end{bmatrix} \begin{bmatrix} u(0) \\ u(1) \\ \vdots \\ u(n-1) \end{bmatrix} \quad (A.2)$$

where $x(j)$, $u(j)$ and $y(j)$ denote the state variable, the system input and output, respectively. And A , B , C and D correspond to state-, control-, output state- and output control-coefficient matrix, respectively. The output $y(j)$ with zero initial state can be defined as

$$\begin{cases} x(j+1) = Ax(j) + Bu(j) \\ y(j) = Cx(j) + Du(j) \end{cases} \quad (j = 0, 1, \dots, n-1) \quad (A.1)$$

The definition of the Toeplitz matrix and the method of identifying the matrix are presented here. A discrete time-invariant system can be described in state-space as

APPENDIX

In this paper, a simplicity and accuracy oriented data-driven control strategy is proposed to improve the servo performance of the developed 6-DOF recycle stage. Compared with model-based control, data-driven approach without the need of dynamics modeling is definitely suitable in terms of the developed recycle stage that possesses high-order and uncertainty dynamics. The design strategy for the three important parts of Ψ_s , C_s and F_s has been discussed in this paper. Furthermore, a data-driven gradient approximation based on the Toeplitz matrix is then developed for updating the parameter vector. Finally, three experiments have been sequentially conducted to establish the overall control system. The comparative results indicate that the performance of the recycle stage has been remarkably enhanced, which validate the effectiveness of the proposed data-driven strategy.

CONCLUSIONS

error of each DOF is attenuated obviously, especially in y -direction. The value of the criterion $J(p)$ before and after optimization is 4.3×10^{-10} and 0.87×10^{-10} , which validates the improvement of the tracking performance. Additionally, due to the nonlinear characteristics, the servo error in the interval of acceleration and deceleration is not symmetrical. To further improve the tracking performance, the adaptive algorithm can be employed to the tuning strategy.

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ACKNOWLEDGEMENTS

Generally, both model-based and data-driven approaches are enabling methods to obtain the Toeplitz matrix. Specifically, model-based approach requires the state-space dynamic model to estimate the Toeplitz matrix, which needs to identify the model by identification. Data-driven approach obtains the Toeplitz matrix through the impulse response experiment, while the output time-series of the system correspond to the Markov parameters.

HYSTERESIS MOTOR DRIVEN ONE AXIS MAGNETICALLY SUSPENDED REACTION SPHERE

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INTRODUCTION

The Attitude and Orbit Control System (AOCS) plays an essential role in the flight control of a spacecraft. This system usually contains a minimum of three reaction wheels (often 4-5 wheels are used for optimization and redundancy [1]). By accelerating the appropriate wheels, the system can produce a zero-mean reaction torque about any axis to the spacecraft, which enables the spacecraft to maneuver on orbit. Meanwhile, the momentum generated by acceleration can be stored in the wheels.[2].

As an alternative to the reaction wheels, the idea of using a single magnetically levitated reaction sphere for satellite attitude control was proposed [3]. In this alternate version, the sphere can be accelerated about any axis by a spherical motor, making the attitude of the spacecraft in all axes controllable by a single device. Due to its symmetry, a sphere can always give the same inertia, independent of its rotational axis. As a result, using a reaction sphere may allow smaller size and mass. In addition, when assisted by magnetic suspension, the mechanical friction can be eliminated, and thus low steady-state power consumption may be possible. This also enables the device to work without lubrication and achieve greater operational lifetime.

The idea of reaction sphere was proposed as early as 1986 [3], however to our knowledge, the current technology is still far from a ready-to-commercialize state. One recent reaction sphere design presents a permanent magnet motor based reaction sphere designed by ESA [4]. This design uses tiled magnets on the rotor surface, which enables simpler angular position sensing and control. However, its complexity of the rotor structure may prevent it from being suitable for small satellites application, and the strength of the rotor limits its maximum rotational speed, which may limit the performance of the actuator.

Among many motor driving principles, the hys-

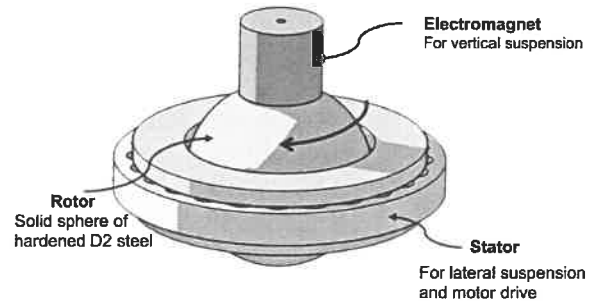


FIGURE 1. Design concept diagram of the 1D-MSRS.

teresis motor is well known for its simple structure, vibration-free operation, self-starting ability, and constant torque production. Another distinct feature of this motor is that its rotor can be made out of a single piece of hard steel, which allows the rotor to stand large stresses and thus makes this motor attractive for high-speed applications. However, to the best of our knowledge, the hysteresis motor has not yet been used in reaction wheels.

Aiming at the dual goal of exploring the design for a magnetically suspended reaction sphere and evaluating the performance of hysteresis motors for reaction wheels application, we focused this project on the development of a magnetically suspended reaction sphere with one-axis hysteresis drive (1D-MSRS). The hardware of the 1D-MSRS demonstrates a solid steel sphere magnetically suspended in all translation directions, and driven by a hysteresis motor about one axis. Figure 1 shows a diagram of the design concept of the 1D-MSRS.

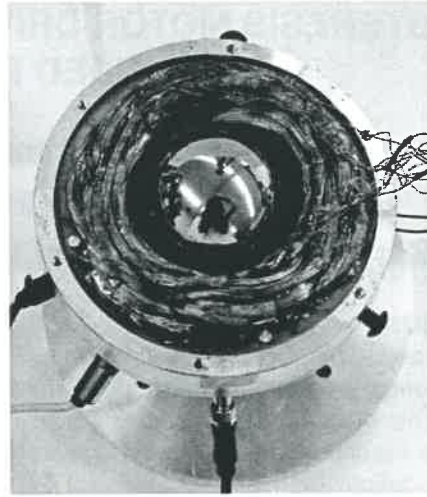
This paper presents the design characteristics and test results of the 1D-MSRS system. For more detailed analysis of the design, modeling, and control of the system see [5].

HARDWARE DESIGN AND INTEGRATION

The 1D-MSRS has a magnetically levitated spherical rotor that can rotate and store momentum about the vertical axis. Figure 2 shows the pho-



(a) Structure and sensors.



(b) stator and rotor.

FIGURE 2. Photographs of the 1D-MSRS hardware.

tographs of the hardware. In the 1D-MSRS system, the rotor is a 54 mm diameter sphere of hardened D2 steel. Four inductive sensors are placed around the rotor to measure the sphere's position in translational degrees of freedom. The rotor sphere is magnetically levitated in the vertical direction by a reluctance actuator placed at its north pole. A stator is arranged around the sphere's line. It serves both for levitating the sphere in the horizontal plane and for torque generation simultaneously via a bearingless motor configuration. A reflective optical tachometer is used for speed detection of the reaction sphere.

The 1D-MSRS consists of several subsystems. They are: a single degree-of-freedom magnetic suspension system for the sphere's vertical suspension, a lateral suspension system for the sphere by mean of the bearingless motor, and a hysteresis motor for driving the sphere about the vertical axis. In the analysis of the 1D-MSRS, these subsystems are considered to be decoupled, that is, the interaction between the subsystems are considered to be negligible. In the following these subsystems are introduced.

VERTICAL SUSPENSION

The spherical rotor in the 1D-MSRS is magnetically suspended in the vertical direction by an electromagnet arranged at the north pole of the sphere.

In order to reduce the DC current in the actuator coil for the sphere's weight compensation, a thin-disk shape permanent magnet is placed in the

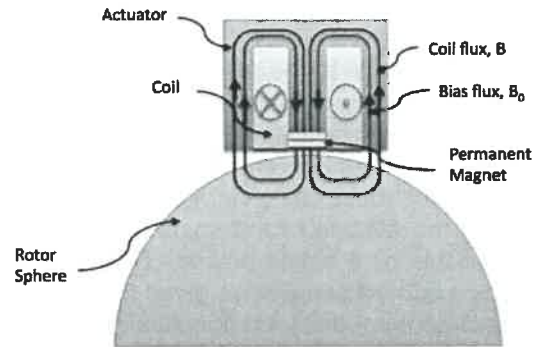


FIGURE 3. A cross section of the biased levitation actuation system

magnetic path of the suspension actuation system to add a bias DC flux. Figure 3 depicts the system for the sphere's vertical suspension.

LATERAL SUSPENSION

In the design of the 1D-MSRS, a bearingless motor is used to achieve a compact design. This motor uses two sets of three-phase windings on a single stator (for 1D-MSRS 4-pole and 2-pole). By correctly configuring and controlling the currents in these motor windings, the machine can generate a torque for spinning as well as radial forces for suspension using a single stator. In the 1D-MSRS system, the 4-pole winding is the motor winding, and the 2-pole winding is used for lateral suspension control.

A dynamic model is built to study the lateral suspension characteristics of the bearingless motor in 1D-MSRS. Here we only present the high level

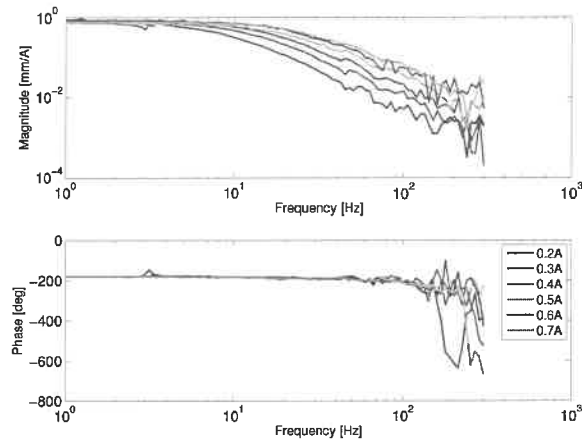


FIGURE 4. Experimentally measured plant frequency response for x -direction sphere suspension with different excitation amplitudes.

result, that is, the transfer function from the suspension winding current i_{2a} to the rotor's radial displacement $x(t)$ is:

$$\frac{X(s)}{I_{2a}(s)} = \frac{K_i}{ms^2 - K_s}. \quad (1)$$

Here m is the mass of the sphere, and the value of K_s and K_i are the negative stiffness [N/m] and the force constant [N/A] of the lateral suspension system respectively. They can be calculated by:

$$K_s = \frac{2}{\pi} \frac{Rl\mu_0 N_4^2 (\sqrt{3}/\sqrt{2}I_m)^2}{g_0^3} [N/m] \quad (2a)$$

$$K_i = \frac{\sqrt{3}}{\sqrt{2}} \frac{2\mu_0 RlN_2N_4}{\pi g_0^2} \left(\frac{\sqrt{3}}{\sqrt{2}} I_m \right) [N/A]. \quad (2b)$$

Here the value I_m is the zero-to-peak current amplitude of the 3-phase current in the 4-pole motor windings. The meaning of the nomenclatures and the detailed derivations of these results are presented in [5].

Equation 2 shows that the both K_s and K_i in the bearingless motor system are varying with the motor winding current amplitude I_m . This result can be verified by measuring the frequency response of the lateral suspension system of the 1D-MSRS under different excitation current amplitude. The measured Bode plots are depicted in Figure 4. This measurement was taken while under closed-loop control of the lateral suspension.

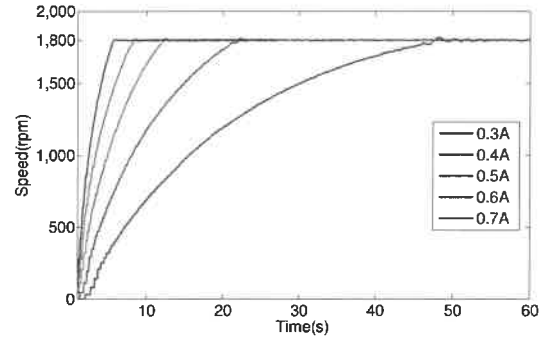


FIGURE 5. Acceleration curves of 1D-MSRS under different excitation current amplitude.

Since the motor excitation amplitude of the reaction sphere/wheel needs to vary according to the torque requirements, the lateral suspension controller needs to be able to stabilize the control loop under all excitation conditions. The detailed controller design for the lateral suspension in 1D-MSRS is presented in [5]. With the controller design, the lateral suspension loop of the reaction sphere kept a constant phase margin of 40° , while the cross-over frequency varies with the motor excitation amplitude.

HYSTERESIS MOTOR

One goal of this project is to evaluate the performance of hysteresis motor for reaction wheels application. A hysteresis motor operates by the magnetic hysteresis effect of its rotor material. Because the magnetization produced in the ferromagnetic material lags behind the magnetizing force, a torque is generated due to the rotor and stator field interaction. Although materials with better hysteresis properties exists, we selected D2 steel for the rotor of 1D-MSRS for the proof of our design.

Experimental data

The start up speed curve of 1D-MSRS is measured under different excitation conditions. Figure 5 presents the acceleration curves of the 1D-MSRS under different amplitudes of excitation current. Data shows that with 0.7 A excitation current, the sphere can reach the synchronous speed of 1,800 rpm within 6 seconds. Also, a starting torque of 8.15 mNm is demonstrated under 0.7 A exciting current.

The maximum synchronous rotational speed that the 1D-MSRS can reach is 200 Hz (12,000 rpm) in our lab. We believe that when running in vacuum the motor has the potential to reach higher speed.

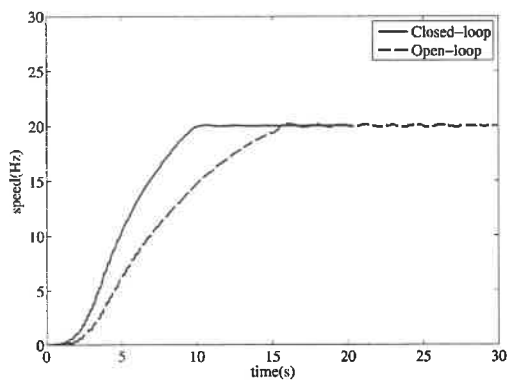


FIGURE 6. A comparison of experimental open-loop speed data and the closed-loop speed data of the 1D-MSRS during starting up.

Speed control of 1D-MSRS

In the speed plot shown in Figure 5, a speed fluctuation slightly above and below the reference speed can be observed when the motor speed first reaches the synchronous speed. This motor dynamics is known as hunting. It is undesirable when a hysteresis motor is used for the development of a reaction wheel or a reaction sphere, as it will introduce vibrations into the spacecraft.

To suppress the motor hunting, a feedback loop on the sphere's rotational speed is designed and implemented in the 1D-MSRS. In this control system, the sphere's speed is measured by an optical tachometer, and the control effort is the current amplitude that we supply to the motor windings. The controller design is introduced in [5] in detail.

Figure 6 shows the measured step response of the reaction sphere's rotational speed under open-loop and closed-loop operations, respectively. Note the hunting speed ripple in the open-loop speed data (blue). Data shows that the designed speed control scheme can effectively suppress the motor hunting and also enables faster acceleration.

CONCLUSIONS AND FUTURE WORK

The design and development of a hysteresis motor driven magnetically suspended one-axis reaction sphere is presented in this paper. Magnetic suspension, bearingless drive and hysteresis motor principles are used in the design for 1D-MSRS. An equivalent circuit model for hysteresis motor is used to analyze the dynamic behavior of the 1D-MSRS, and a speed control loop is built for the system to suppress hunting. The 1D-MSRS can run up to 200 Hz (12,000 rpm) with the existence of air drag, and a starting torque of 8.15 mNm is

generated with 0.7 A excitation current amplitude.

Future work should consider the design and development of a three-axis magnetically suspended reaction sphere (3D-MSRS). A brief discussion of the possible motor concepts and magnetic pole configurations for a 3D-MSRS is presented in [5]. The performance of the 1D-MSRS demonstrates that hysteresis motor has the potential for the circumstances where quiet operation and high speed are needed. We believe that this motor concept is promising for high speed, low vibration reaction wheel applications.

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LARGE-STROKE PIEZO-ACTUATED PLANAR MOTOR FOR NANOPositionING APPLICATIONS

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INTRODUCTION

High precision planar motion systems are widely used in modern industrial and scientific applications, such as semiconductor manufacturing, precision machining, etc. Many of these applications require sub-micron to nanometer positioning resolution and accuracy as well as a large range of motion. Conventional multi-degrees-of-freedom (multi-DOF) stages are constructed by stacking multiple single-DOF stages together, which are relatively easy to construct and control. However, such stages are typically too bulky for fast motion response. They are also prone to errors due to error accumulation and misalignment between motion axes.

To overcome the above drawbacks of conventional multi-DOF stages, various planar motion systems have been developed, including electromagnetic planar motors [1], ultrasonic planar motors [2], stick-clamping planar motion stages [3], and stick-slip planar motion stages [4]. These planar motion systems use a single mover and control the multi-DOF motion directly. Therefore, they do not have the drawbacks of the conventional systems. However, they also have their own shortcomings. Electromagnetic planar motors require complex dynamic motion control. Ultrasonic planar motors require complex dynamic analysis of the resonator and are still immature for real applications. Stages based on the stick-clamping method using piezo actuation have a limited velocity because the clamping action in each step stops the stage. Stages based on the stick-slip effect have a limited frictional driving force and suffer from severe wear due to the existence of slip.

To overcome some of the drawbacks of the planar motion systems described above, a novel large-stroke piezo-actuated planar motor is proposed, which has the following features:

- 3-DOF ($XY\theta_z$) positioning control
- Large stroke (only limited by the base size)
- Large rotation angle (up to 360°)

- Nanometer level positioning resolution
- Multiple positioning modes (high resolution or high speed)
- Omni-directional mobility
- 6-DOF final position adjustment
- High static stiffness and good stability

The following sections describe the design, operation modes, and control of the proposed planar motor and present some preliminary experimental results to show its feasibility.

DESIGN OVERVIEW

Figure 1 shows the proposed planar motor, which has four piezo legs located at each corner. The size of the planar motor is 120×120 mm. A vacuum-preloaded air bearing is positioned in the center, which provides supporting and clamping forces for the planar motor. Two 3-DOF piezo legs, each providing XYZ motion control at the tip of the leg, are mounted at two diagonally opposite corners. Two 1-DOF piezo legs providing Z motion control are mounted at the other two corners. The planar motor has

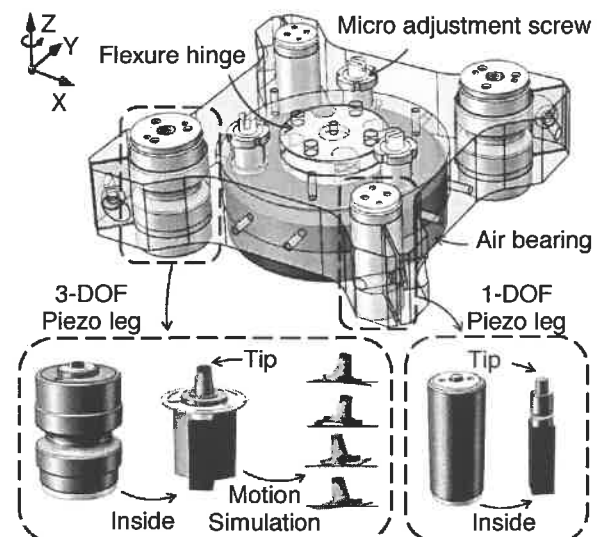


FIGURE 1. Design of the piezo-actuated planar motor

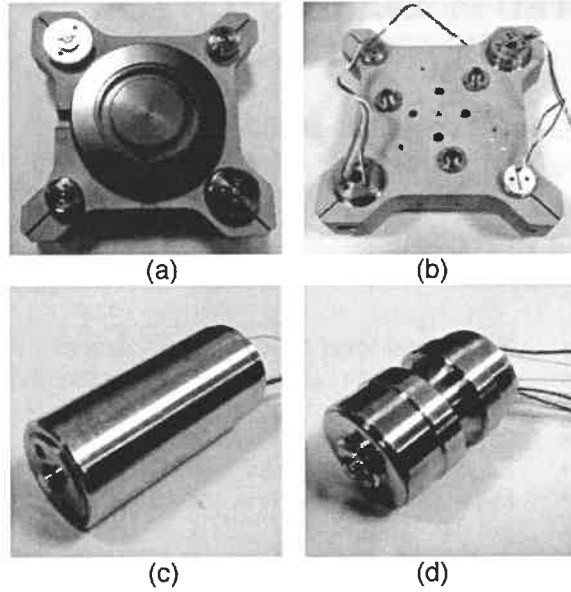


FIGURE 2. Photos of the planar motor. (a) Top view. (b) Bottom view. (c) 1-DOF piezo leg. (d) 3-DOF piezo leg.

three DOFs ($XY\theta_z$) and omni-directional motility with four piezo legs operating cooperatively under different operation modes [5]. Figure 2 shows some photos of the planar motor.

OPERATION MODES

With the four piezo legs working cooperatively, the planar motor can operate under three different operation modes: the skiing mode, the stepping mode, and the tilting mode (Figure 3). In the skiing mode, the two 3-DOF legs move like ski sticks in high frequency to drive the planar motor while the two 1-DOF legs are retracted and not used. Due to the frictionless condition provided by the air bearing, the planar motor can move in high speed under this mode. In the stepping mode, the 3-DOF legs move step by step in a low frequency. The 1-DOF legs extend to clamp on the base when the 3-DOF legs are retracted in between the steps. Due to this clamping effect, the stepping mode provides more accurate position control compared to the skiing mode. In the tilting mode, all four legs clamp on the base. Fine position adjustment of the mover in 6-DOF is achieved through the tilting of the motion head of the 3-DOF legs and extension/contraction of the 1-DOF legs. This operation mode provides the most stable and highest resolution position control. The above three operation modes can also be combined to achieve both speed and accuracy in position control.

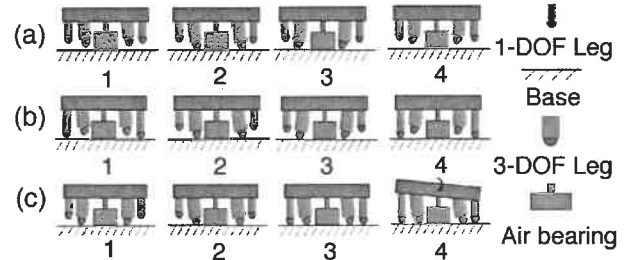


FIGURE 3. Three operation modes. (a) Skiing mode. (b) Stepping mode. (c) Tilting mode.

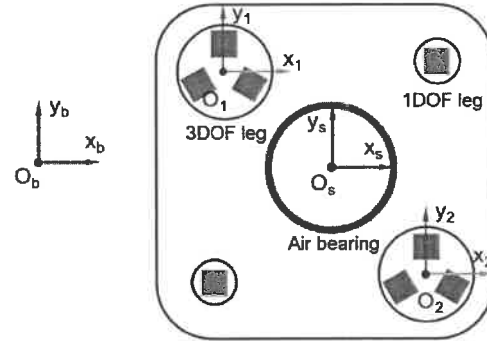


FIGURE 4. Definition of the coordinate systems

MOTION CONTROL

This section describes the control algorithms of the four piezo legs under three operation modes. Four coordinate systems O_b , O_s , O_1 , O_2 are defined on the base, mover, 1st, and 2nd 3-DOF legs respectively as shown in Figure 4. Assume that the desired velocity of the mover with respect to the base coordinate system O_b is

$$\mathbf{V}_s^b = [\dot{x}_s^b \ \dot{y}_s^b \ \dot{\theta}_z^b]^T. \quad (1)$$

Then the desired velocity of the i th 3-DOF leg with respect to its own coordinate system is

$$\mathbf{V}_i^i = \mathbf{J}_i(\theta_z) \mathbf{V}_s^b, \quad (2)$$

where the Jacobian is

$$\mathbf{J}_i(\theta_z) = \begin{bmatrix} \cos \theta_z & \sin \theta_z & -y_i^s \\ -\sin \theta_z & \cos \theta_z & x_i^s \end{bmatrix}. \quad (3)$$

3-DOF Leg Control

The motion head in each 3-DOF leg is controlled by three piezo actuators to provide 3-DOF motions in a spherical coordinate system (δ , ϕ , and Z), as illustrated in Figure 5. The angle δ defines the motion plane of the 3-DOF leg, in which the motion head moves. Angle δ can be dynamically controlled to provide omni-directional mobility of the mover. It can be calculated using the velocity vector $\mathbf{V}_i^i$ in Eq. (2). Rotation angle ϕ and linear displacement Z define the position of the tip of the motion head in the

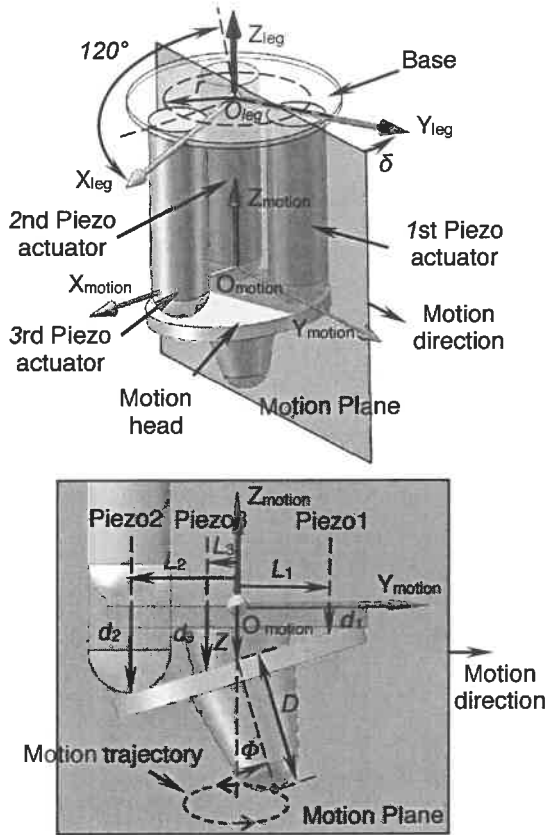


FIGURE 5. Definition of parameters and coordinate systems of the 3-DOF piezo leg

motion plane. The displacement of the j^{th} piezo actuator d_{j_3DOF} in the 3-DOF piezo leg can be obtained as

$$d_{j\_3DOF} = -Z - r \cos(120^\circ(j-1) + \delta) \Phi, \quad (4)$$

where r is the radius of a circle along which the three piezo actuators are evenly positioned.

Control Algorithm for the Skiing Mode

In the skiing mode, the tip of each 3-DOF piezo leg moves along an ellipsoidal trajectory repetitively in high frequency to drive the mover in high speed, as shown in Figure 5. This ellipsoidal motion can be generated by controlling Z and Φ as follows:

$$Z = A_z \cos(2\pi f t + \pi/2), \quad (5)$$

$$\Phi = A_\phi \cos(2\pi f t), \quad (6)$$

where A_z and A_ϕ are the height and span of each motion step respectively. There is a phase difference of $\pi/2$ between Z and Φ , which is needed in order to generate the ellipsoidal motion trajectory. The motion frequency f can be used to adjust the motion speed. Substitute Eqs. (5) and (6) into Eq. (4), we can derive the motion of each piezo actuator as

$$d_{j\_3DOF} = A_j \cdot \cos(2\pi f t + \theta_j), \quad (7)$$

where A_j and θ_j have the following forms

$$A_j = \sqrt{r^2 \cos^2(120^\circ(j-1) + \delta) A_\phi^2 + A_z^2}, \quad (8)$$

$$\theta_j = \text{atan2}(-A_z, -r \cos(120^\circ(j-1) + \delta) A_\phi). \quad (9)$$

Control Algorithm for the Stepping Mode

In the stepping mode, all four piezo legs work cooperatively to drive the mover step by step. The control of the 3-DOF legs is the same as that in the skiing mode but with a relatively low driving frequency (less than 50 Hz). The motion of the 1-DOF legs has a phase difference of π compared to the Z motion of the 3-DOF legs, which means that the 1-DOF legs can extend to clamp the mover when the 3-DOF legs are lifted up. The motion equation of the 1-DOF legs is

$$d_{i\_1DOF} = A_i \cdot \cos(2\pi f t - \pi/2). \quad (10)$$

Control Algorithm for the Tilting Mode

In the tilting mode, the 3-DOF and 1-DOF legs tilt, extend, or contract to fine adjust the position of the planar motor in 6-DOF. Given the desired position adjustment vectors $\Delta P1_s^b$ and $\Delta P2_s^b$, the displacement of the j^{th} piezo actuator in j^{th} 1-DOF leg and j^{th} 3-DOF leg can be derived as

$$d_{i\_1DOF} = [1 \quad y_{i\_1DOF}^s \quad -x_{i\_1DOF}^s] \Delta P2_s^b, \quad (11)$$

$$d_{i\_j\_3DOF} = -Z_i - r \cos(120^\circ(j-1) + \delta_i) \Delta \Phi_i, \quad (12)$$

where

$$\Delta P1_s^b = [\Delta x_s^b \quad \Delta y_s^b \quad \Delta \theta_z^b]^T, \quad (13)$$

$$\Delta P2_s^b = [\Delta z_s^b \quad \Delta \theta_x^b \quad \Delta \theta_y^b]^T, \quad (14)$$

$$\delta_i = \text{atan2}(J_i(\theta_z) \Delta P1_s^b), \quad (15)$$

$$\Delta \Phi_i = -|J_i(\theta_z) \Delta P1_s^b| / D, \quad (16)$$

$$Z_i = [1 \quad y_{i\_3DOF}^s \quad -x_{i\_3DOF}^s] \Delta P2_s^b. \quad (17)$$

EXPERIMENTAL SETUPS

For closed-loop motion control of a planar motor, a multi-DOF position sensor is necessary to provide position feedback. In this research, two novel active planar encoders were developed, one with a high resolution but a small measurement range [6] and the other with a lower resolution but a large measurement range.

Figure 6 shows the experimental setup with the high-resolution active planar encoder, which is used to evaluate the position control resolution of the planar motor. A mover supported by an air bearing is placed on a granite base. An iPhone 4S with a uniform white image on the screen is mounted on top of the mover. A camera with a

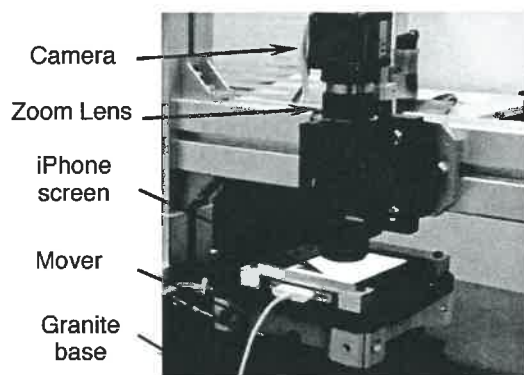


FIGURE 6. Experimental setup with a high-resolution active planar encoder

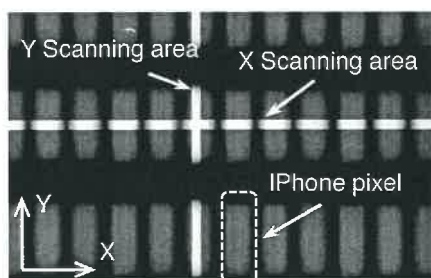


FIGURE 7. Pixel image of the iPhone screen

50x objective lens is used to capture images of the screen pixels (see Figure 7). A phase correlation algorithm is used to calculate the position information from the images with a resolution of 6 nm. Even though this encoder has a high linear displacement measurement resolution, it has a small measurement range. Besides, it does not measure rotation angles and its measurement speed is limited to 12 Hz due to the large exposure time necessary for capturing good quality images.

Figure 8 shows the experimental setup with a large-range active planar encoder, which is used to evaluate the position control performance of the planar motor in a large range of motion. This encoder uses two cameras, each with a compact low-distortion 1x lens. The iPhone 4S mounted on top of the mover is programmed to display two perpendicular sinusoidal fringe patterns with a pitch of 0.78 mm. Each camera captures a sinusoidal fringe image from the iPhone screen. Three sinusoidal signals $X1$, $X2$, and $Y1$ are obtained from three scanning areas of the two captured images, as shown in Figures 8(b) and 8(c). The phase of each sinusoidal signal is calculated using a novel phase estimation algorithm, which will be discussed in a future publication. The phase of the sinusoidal

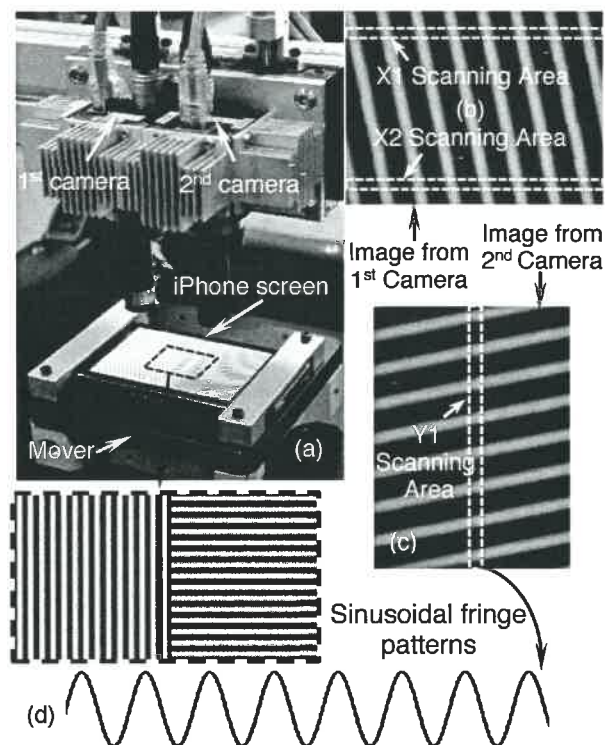


FIGURE 8. Experimental setup with a large-range active planar encoder. (a) Planar motor with the encoder. (b) $X1$ and $X2$ scanning area for calculating X and θ_z . (c) $Y1$ scanning area for calculating Y . (d) Sinusoid signal of $Y1$.

signal change periodically when the planar motor moves. Therefore, a phase unwrapping algorithm is needed to convert the phase information into its corresponding position signal. The phases of $X1$ and $X2$ are used to calculate position X and angle θ_z and the phase of $Y1$ is used to calculate the position Y of the mover.

The linear measurement range of this encoder depends only on the display size. The rotation measurement range is up to $\pm 45^\circ$, but can be extended to $\pm 180^\circ$ if the two fringe patterns on the screen is switched dynamically. The linear measurement resolutions along X - and Y -axis are 50 nm and 500 nm respectively and the measurement accuracy in a range of 40 mm is $\pm 0.6 \mu\text{m}$. The angular measurement resolution of θ_z is 2.2 arcsec. The angular measurement accuracy of the encoder will be evaluated in the future. Since a shorter camera exposure time can be used with this setup, a higher measurement speed of 100 Hz can be achieved. Besides, the absolute positions of the mover can also be measured with a coding method.

EXPERIMENTAL RESULTS

The experimental setup with a high-resolution active planar encoder is used to test the static stiffness and positioning resolution of the planar motor. The experimental setup with a large-range active planar encoder is used to evaluate the performance of the planar motor under the stepping mode and skiing mode. The closed-loop position control of the mover is done with a PID controller.

Static stiffness test

The static stiffness of the planar motor under the tilting mode is tested using a tension meter, as shown in Figure 9(a). The result shows that the planar motor has a static stiffness of $1.33 \text{ N}/\mu\text{m}$. Figure 9(b) shows the mover clamped to the erected base without falling down, which demonstrates that the planar motor has a large static clamping force.

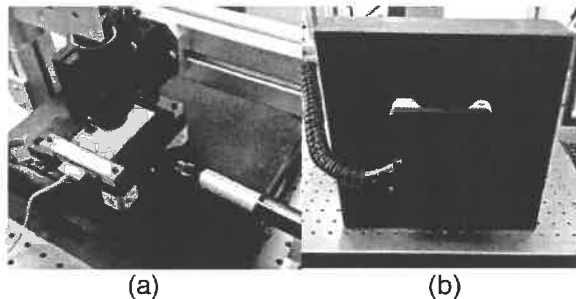


FIGURE 9. Static stiffness and clamping force tests. (a) Stiffness test. (b) Clamping force test.

Tilting mode test

A step test along the X-axis has been conducted using the experimental setup with a high-resolution active planar encoder. The operation mode is the tilting mode and the step size is 100 nm. Figure 10 shows the result. The standard deviation of the step test result is approximately 6 nm, which is the resolution limit of the encoder. This means that the positioning resolution of the planar motor is currently limited by the encoder used. In the future, a laser interferometer will be used to test the ultimate positioning resolution of the planar motor. A circular tracking test with a diameter of $10 \mu\text{m}$ was also conducted and the result is shown in Figure 11. This result demonstrates the capability of the planar motor in simultaneous XY position control at the nanometer level.

Stepping mode test

The experimental setup with a large-range active planar encoder is used to evaluate the

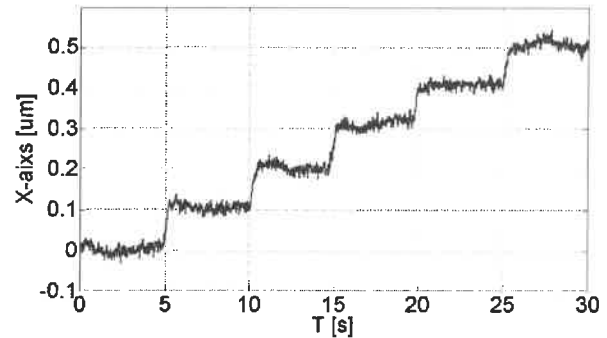


FIGURE 10. 100-nm step response

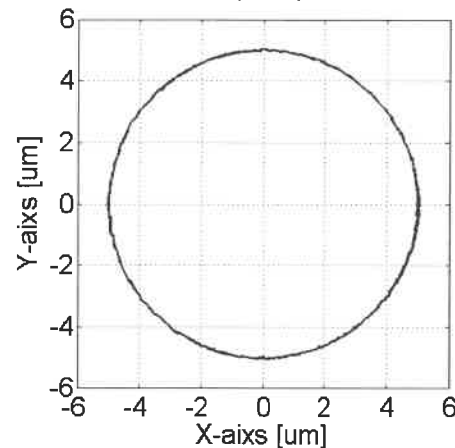


FIGURE 11. Circular tracking performance

position control performance of the planar motor in a large range. The planar motor is controlled to follow a special motion trajectory with the linear position of the mover following a circle and the angular position θ_z following a sinusoidal curve with an amplitude of 5° . The result of this tracking test is shown in Figure 12. The peak-to-peak deviation of the circular motion is approximately $\pm 1 \mu\text{m}$, which is comparable to the noise level of the planar encoder. The peak-to-peak deviation of the rotational motion is approximately $\pm 20 \text{ arcsec}$.

Skiing mode test

The speed of the planar motor can be adjusted by changing the driving frequency and the step size of the 3-DOF piezo legs. Figure 13 shows the relationship between the driving frequency and the planar motor speed for different step sizes. Obviously, the larger the step size and the higher the driving frequency is, the higher the speed of the planar motor is.

CONCLUSION

In this research, a novel large-stroke planar motor driven by four piezo legs was developed

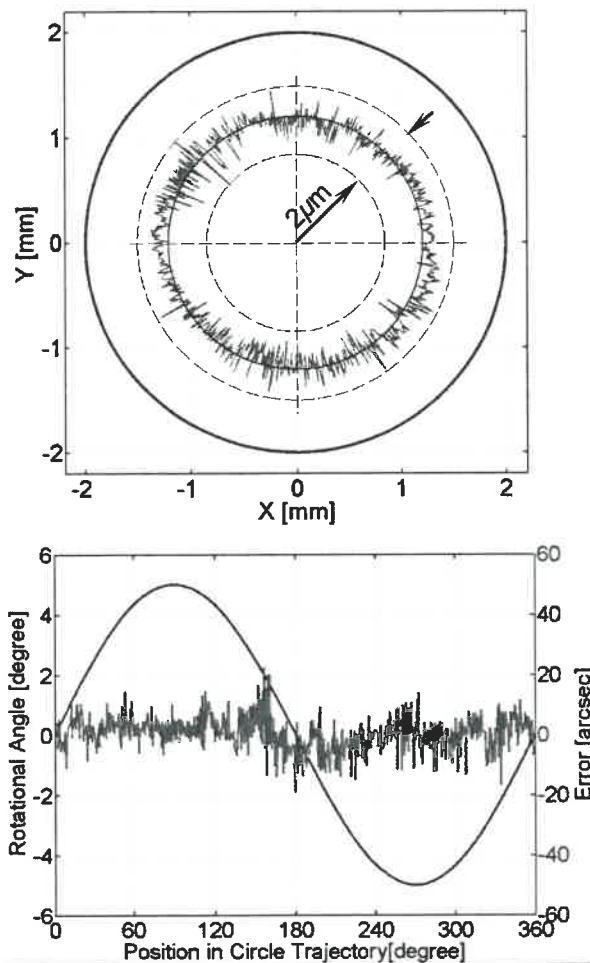


FIGURE 12. Large range motion performance

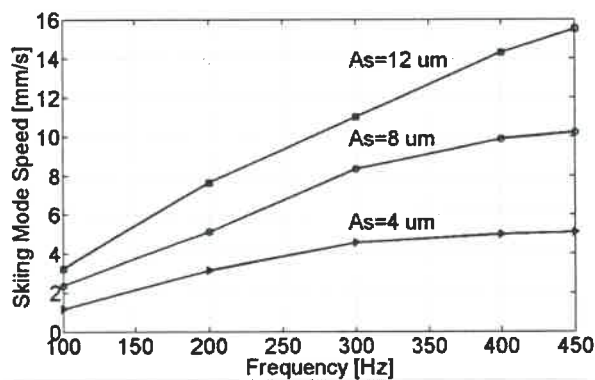


FIGURE 13. Planar motor speed vs. driving frequency under the skiing mode with different step sizes.

for nano-positioning applications. Its linear range of motion is limited only by the size of the base or the measurement range of the encoder for a closed-loop system and its angular range of motion is up to 360°. There are three possible operation modes: the skiing mode, the stepping

mode, and the tilting mode. The skiing mode provides the highest positioning speed with a relatively low resolution. The stepping mode provides a high resolution, but a relatively slow speed. The tilting mode offers the finest position control in 6-DOF. The above operation modes can also be combined to achieve both speed and accuracy in position control. For position feedback, novel active planar encoders were developed and integrated with the planar motor for closed-loop position control. Experimental results demonstrated that the proposed planar motor could simultaneously achieve nanometer level positioning resolution and large ranges of linear and angular motions.

Currently, the performance of the planar motor is limited by the performance of the encoder used. As a future work, we plan to further improve the measurement resolution and speed of the active planar encoder to explore the true potential of this unique planar motor for nano-positioning applications.

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TRACKING CONTROL PERFORMANCE OF ULTRAHIGH ACCELERATION AND HIGH VELOCITY LINEAR DRIVE MECHANISM USING LEARNING CONTROL METHOD

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INTRODUCTION

In precision industrial machines, for example, machine tools and measuring machines, their motion characteristics greatly influence the machine performances. High tracking accuracy and high responsiveness are important motion characteristics for the machines. These characteristics essentially depend on those of linear drive mechanisms. The acceleration and the velocity are often used as important indicator of the responsiveness and their required levels are becoming strict increasingly [1].

For the requirements, linear motors are widely used in recent years. They are direct-drive and have the advantages of small driven inertia, no speed limitation by the mechanical structure, ease to control. Particularly the linear motors are generally used for the acceleration higher than 19.8m/s^2 ($\approx 2\text{G}$) and the velocity higher than 2m/s . In the market, there are linear motors of which maximum acceleration is about 40G [2].

The purpose of this research is to realize precision and high speed motion with an ultrahigh speed linear drive mechanism which can move at the acceleration higher than 70G and the velocity higher than 10m/s . For this purpose, the ultrahigh-acceleration and high-velocity linear drive mechanism has been designed and the driving performances of the prototypes have been examined [3]. The prototypes have shown the acceleration higher than 70G and the velocity higher than 10m/s . However the tracking error at high speed was too large for high precision industrial machines. This paper describes precision and high speed tracking control of the ultrahigh-acceleration and high-velocity linear drive mechanism.

ULTRAHIGH-ACCELERATION AND HIGH-VELOCITY LINEAR DRIVE MECHANISM

Construction of the Prototype

Figure 1 shows a prototype of ultrahigh-acceleration and high-velocity linear drive

mechanism. This linear drive mechanism includes a moving type permanent magnet linear synchronous motor (MPM LSM). The MPM LSM has a stator comprising two electromagnet (EM) lines and a mover including permanent magnets (PMs). The length of stator is 2.03m . The EM lines are located on both sides of the mover.

Figure 2 shows the prototype mover. The length and weight of the mover are 440mm , 4.63kg respectively. The mover is supported by linear ball guide including two guide rails and four carriages. The displacement of the mover is

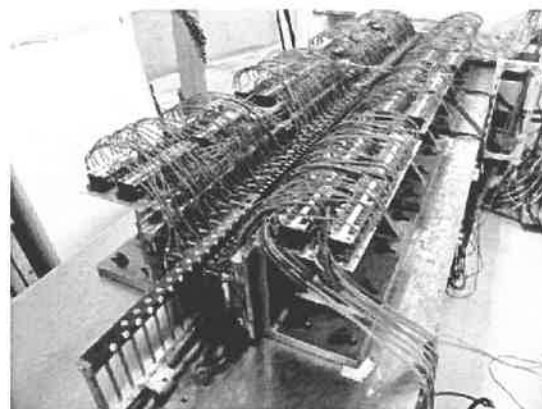


Fig.1 Prototype of ultrahigh-acceleration and high-velocity linear drive mechanism

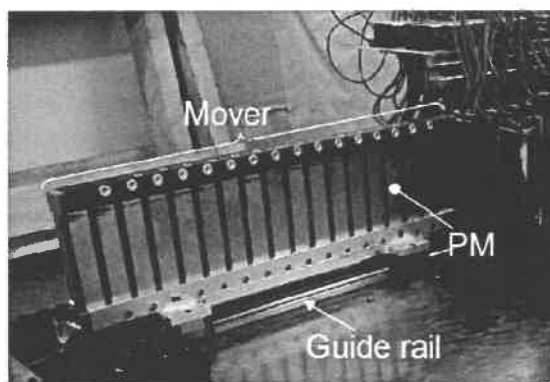


Fig.2 Prototype mover

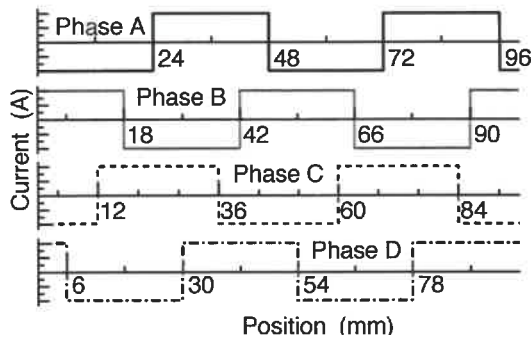


Fig.3 Driving current command signal waveforms

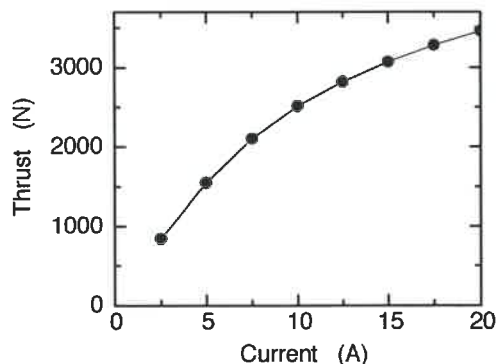


Fig.4 Static thrust characteristic

measured using a laser position transducer (ZLM series from JENAer Meßtechnik GmbH. Resolution: 79.1 nm). The maximum applied current to the EMs is 20A and the maximum voltage is 280V.

Driving Characteristic

The prototype MPM LSM is driven by four current commands which have rectangular waveforms. The waveforms are shown in Fig.3. The current commands are inputted to pulse width modulation (PWM) amplifiers for driving the mover. The phases of the commands depend on the position of the mover. In the control system described later, the controller calculates the suitable amplitude of the waveforms.

The prototype can move at the acceleration higher than 779m/s^2 ($\approx 79.4\text{G}$) and at the velocity higher than 11.1m/s when the applied signals of which amplitudes are 20 A are inputted. The acceleration and velocity are much higher than those of conventional high speed linear drive mechanisms.

The MPM LSM has many nonlinear elements and shows significant nonlinear characteristics.

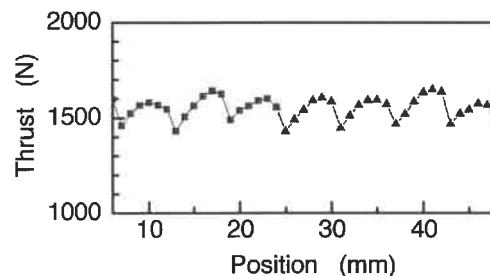


Fig.5 Relationship between mover position and thrust (the command signal amplitude: 10A)

As shown in [4], the coil current responses in the EMs show that the electrical characteristics of the MPM LSM are nonlinear. The relationship between the applied current to the coils and the generated thrust force is also nonlinear. It is influenced by the current actions of neighboring coils, the magnetic saturation and magnetic leakages. The static thrust characteristic is shown in Fig.4. The thrust characteristic is influenced by the mover position because of the combination of PM and cored EM. Figure 5 shows an example of the relationship between mover position and thrust. In the figure, the signal amplitude was 10 A. Friction force is caused in the linear guides.

CONTROL SYSTEM DESIGN

Previous Control System and its Problem

The MPM LSM has distinct nonlinear characteristics as described in the previous section. The characteristics make it difficult to

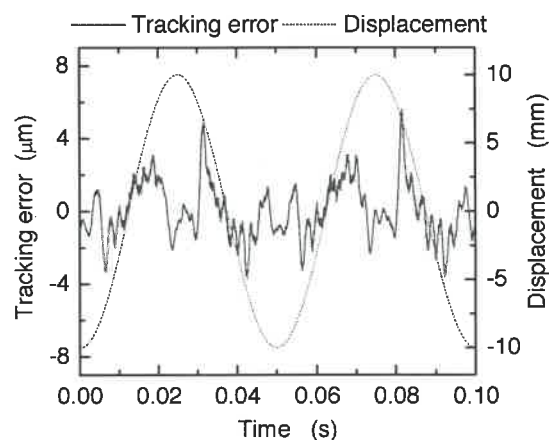


Fig. 6 Experimental tracking error on the sinusoidal reference motion without the learning control element (Frequency: 20 Hz, Amplitude: 10 mm)

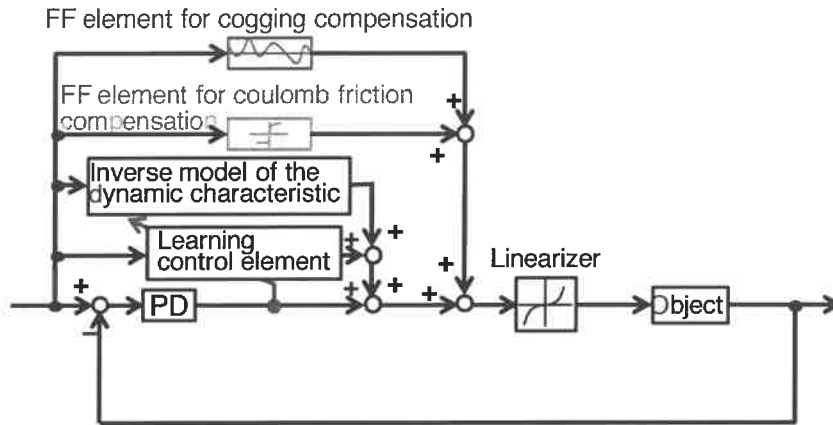


Fig.7 Improved control system with a learning control element

construct an exact dynamic model for the control system design. Although the dynamic model of the MPM LSM in [4] can be used for simulating the macroscopic motion, it is unsuitable for the detailed controller design.

In the previous study, the controller for precision motion of the MPM LSM has a two-degree-of-freedom (2DOF) compensator which consists of a nonlinear PID element and a feedforward (FF) element determined from simplified inverse model. In addition, control elements such as FF elements for compensating the cogging force and the friction force, a disturbance observer were used to suppress the bad effect of the nonlinear characteristics and unknown error factors. The observer was designed based on the simplified dynamic model of the MPM LSM. The controller provided the low tracking errors on the low frequency responses. However the error increased with an increase of the frequency. Figure 6 shows the tracking error of the prototype MPM LSM with the sinusoidal input of which the amplitude and the frequency are 10mm and 20Hz. The maximum tracking error was larger than 5 μ m.

Improvement of Controller

It is well-known that the use of the detailed inverse model as a FF element is effective to improve the system performance at high frequency motion. However, as described above, it is impractical to use a detailed dynamic model of the MPM LSM in controller design.

In contrast, a learning control method does not request the exact dynamic model in the design and it is easy to use for mechanisms which have significant nonlinear characteristics.

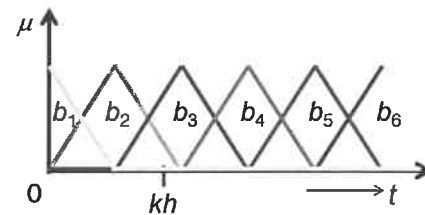


Fig.8 Basis function used in the learning control

Figure 7 shows the improved control system with the MPM LSM for high frequency motion. In the control system, the learning control is used as a FF element [5, 6]. The learning control element is modified using the output signal of the feedback compensator. The basic algorithm for the learning control is as follows:

$$u_F^j(kh) = u_F^{j-1}(kh) + \gamma v^{j-1}(kh)$$

where j denotes the j th repetitive operation, h is the sampling period, k is the time index, $u_F^j(kh)$ is the output of the learning control element at the time of kh , γ is the learning gain, and $v^{j-1}(kh)$ is a value calculated from the used learning algorithm.

The output of the feedback compensator is filtered by the function called the "basis function," and $v^j(kh)$ is calculated. Figure 8 illustrates the basis function used in the learning control. One basis function has one triangular part in a certain period, and the function in the other period equals zero. There are plural basis functions in each movement period. Two triangular parts of the basis functions having adjacent numbers overlap each other. The learning control parameters such as the period of one triangular

part in a basis function and the learning gain is adjusted to suppress the tracking errors.

The improved control system is almost similar to the combination of the previous control system and the learning control element. However the improved control system does not include the integral element and the disturbance observer since their elements degrade the performance of the system including the learning control element.

EXPERIMENTAL RESULTS

Using the controller shown in Fig.7, the sinusoidal response of the prototype MPM LSM was measured. Figure 9 shows the tracking error of the prototype MPM LSM with the sinusoidal input of which the amplitude and the frequency are 10mm and 20Hz, respectively. The tracking error with the learning control element is smaller than $2\mu\text{m}$. The tracking error of the control system with the learning control element was about one third of the error without the learning control. In the figure, the tracking error of the control system with the disturbance observer is shown. The comparison result suggests that the use of the disturbance observer deteriorates the learning control effect on the suppression of the tracking error.

CONCLUSION

In this paper, the control system for precision and high response tracking control of the ultrahigh-acceleration and high-velocity linear drive mechanism and its tracking performance was introduced. For the mechanism, the controller comprising the 2DOF compensator, the FF compensator for cogging and friction force, and the learning control element was designed and implemented to the prototype MPM LSM. The experimental result shows that the learning control element is effective to reduce the tracking errors and that the controller makes the tracking error on the sinusoidal response of which the amplitude and the frequency are 10mm and 20Hz was $1.62\mu\text{m}$.

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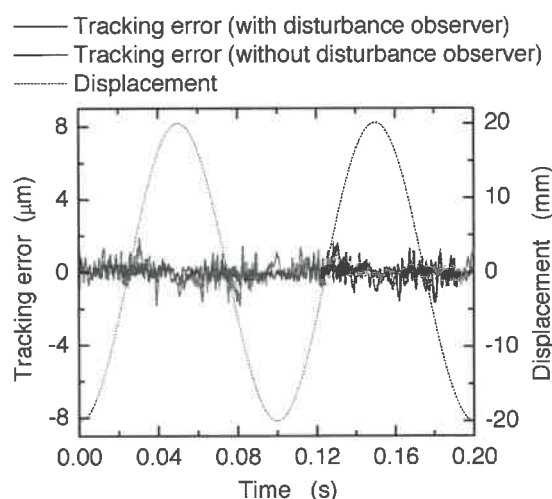


Fig. 9 Experimental tracking error on the sinusoidal reference motion (Frequency: 20 Hz, Amplitude: 10 mm)

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THE DESIGN AND DEVELOPMENT OF A NOVEL 'TWIN TURRET' MACHINING PLATFORM.

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INTRODUCTION

Cranfield Precision has developed a new concept for high precision machining. The classical orthogonal machine tool platform is capable of supporting traditional material removal processes. More recently, ideas such as the Hexapod and the Tetraform [1] have demonstrated benefits for some processes and in the case of the Hexapod style machines, even to find applications outside metal removal in low inertia, high speed metrology [3]. The concept described here leverages capabilities intrinsic to modern control systems to coordinate two rotary axes and a short linear axis in a 'Twin Turret' design which provides a common platform for multiple machine configurations.

MACHINE DESIGN CONCEPT

Cranfield Precision was challenged to find a radical machine configuration that could deliver a highly stiff, thermally stable foundation for a wide range of machine systems, particularly the optics industry for grinding spheric, aspheric and free-form surfaces. Cranfield Precision's experience in the optic industry over many years shows that ductile regime grinding [3] delivers considerable process advantages over conventional glass grinding. This mode of operation is difficult to support unless the machine system is of the highest quality.

The machine uses a unique combination of rotary and linear axes to produce relative motion (both position and angle) between tool and workpiece over a swept working area.

The potential benefits of using rotary bearings rather than linear bearings to provide the primary motion between tool and component has been well known. There have been test machines developed that use only rotary axes [4]. It is however, the addition of the linear infeed axis on the Twin Turret machine that differentiates it from previous machines and has been enabled the

production of very high quality ground surfaces in commercially viable cycle times.

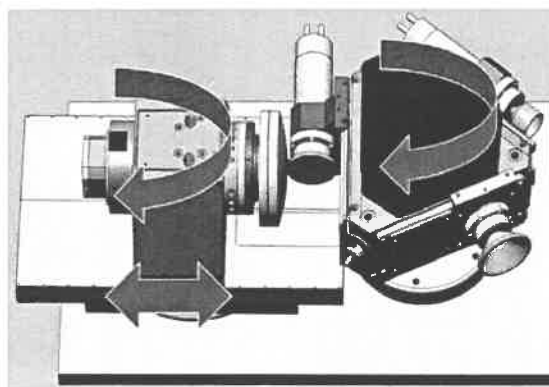


Figure 1: Machine axes concept.

The two rotary axes are rigidly mounted a fixed center distance apart from each other. In order to traverse the tool across components of up to 400mm diameter, the rotary axes need only to rotate by around 20° each and it is possible to align the 'best' segment of the rotary encoders to the critical 20° of motion. It has thus been demonstrated that by using rotary encoders specified with 1 arc sec accuracy over 360° it is possible to reduce the effective error over the 20° working range to 0.1 – 0.2 arc sec, resulting in an uncorrected traverse position error between tool and component of around 0.6µm.

The linear axis is used to control the depth of cut and profile shape of the component being machined. It is a simple procedure during machine build to error correct the linear infeed axis such that the correct profile is followed. It is only necessary to carry out the error correction in one location between the two rotary axes (see fig 2). This is because the center distance between the turrets is fixed and the linear axis is always perpendicular to the 'virtual' linear axis being corrected.

The correction data is thus valid for any position of the linear axis. Fig 2 shows how a Zerodur straight edge can be used to error correct the interpolated straight line motion between tool (here the probe) and component (here the straight edge). The intrinsic error before correction was in this case $200\text{ }\mu\text{m}$ and increases as the two rotary axes move towards the opposite extremes of their 20° travel range. After error correction, the straight line motion error was reduced to $0.3\text{ }\mu\text{m}$.

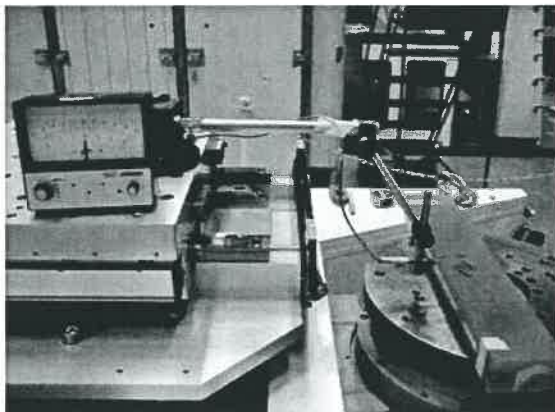


Figure 2: Proof of concept test machine straightness calibration set up.

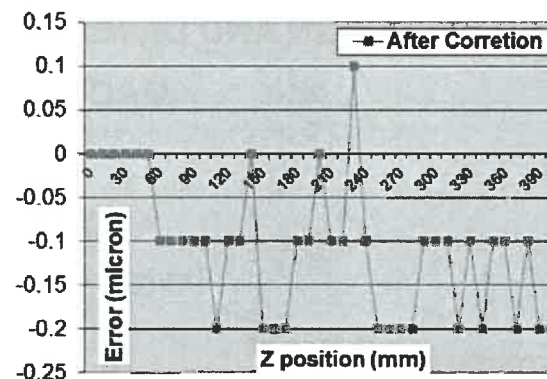
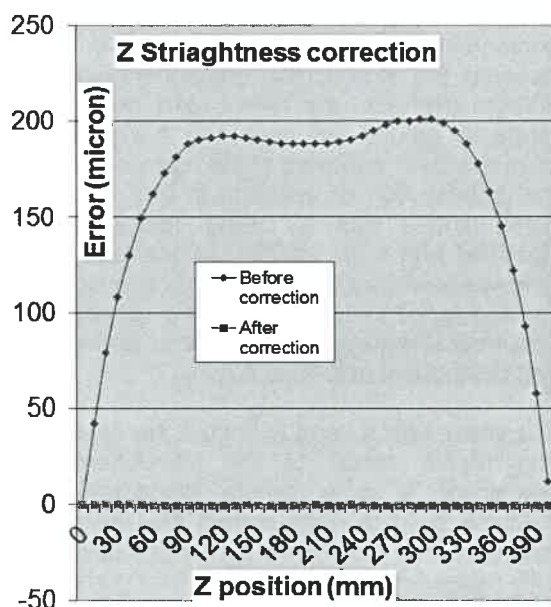


Figure 3: Proof of concept machine straightness, before and after error correction.

Thermal stability

This is a major problem for all machine tools. Conventional grinding machines have large, variable thermal loops that can produce significant movement between wheel and component.

The time taken to grind optical quality surfaces rises exponentially as the specifications for form error, surface finish and sub-surface damage become more challenging and increase part processing time. Larger parts can take days to finish. Such demands dictate a total focus on the maintenance of thermal stability in the presence of coolant, process and ambient thermal influences.

The traditional machine design suffers from a constantly changing coolant return path as the grinding wheel carriage moves along the linear axis. The heat from the grinding process is transferred to different sections of the machine bed, resulting in constantly variable machine distortions. The Twin Turret design enables a simple non-contacting labyrinth seal, making the machine base almost immune to such distortions.

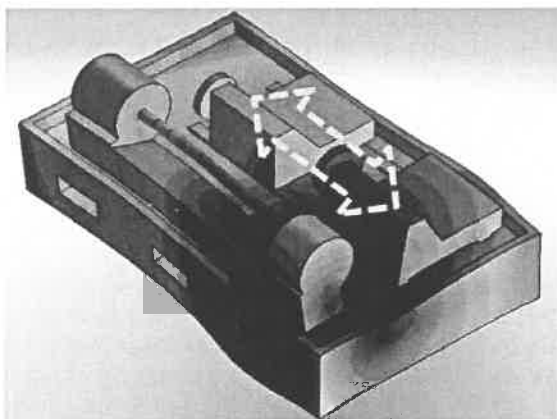


Figure 4: Conventional machine thermal distortion.

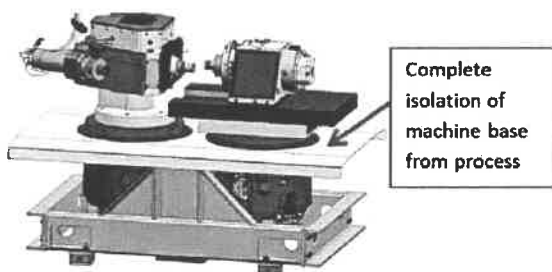


Figure 5: Twin Turret machine base isolation.

Loop stiffness

In conventional machines using stacked linear axes, the bearing interfaces are more compliant than the machine's base material. The target was to use the minimum number of bearing interfaces in the machining loop, thus minimizing compliance. In the optics industry, a common requirement is to achieve 'tool normal' operation of the grinding wheel. A point is selected in the wheel tip which is considered to be a datum that is locked to the cutting point on the workpiece. This method ensures surface texture consistency and reducing the impact of grinding wheel form errors.

Achieving tool normal operation is conventionally achieved by mounting the wheel spindles upon rotary axes that in turn are mounted upon stacked linear axes. Each interface reduces the stiffness of the machine which is counter to the necessity for extremely high stiffness that is needed to support ductile regime grinding. High machine stiffness is essential to maintain

consistent and high quality surface finish and to minimize wheel wear.

The new machine base is effectively two highly stiff ($>10\,000\text{ N}/\mu\text{m}$ radial and axial) rotary hydrostatic bearings. Conventional machines use linear axes to provide the primary motion between tool and component, the machine base has to resist machining forces in bending. The Twin Turret machine's turrets are bolted together via a solid base plate resisting the machining forces in tension rather than in bending.



Figure 6: Rotary turret cross section.

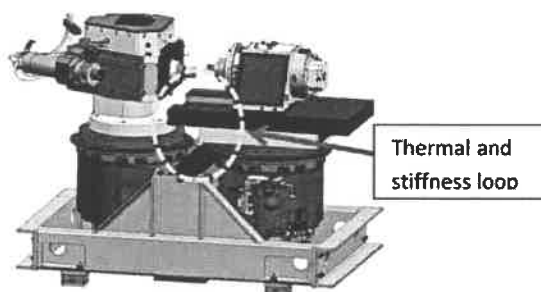


Figure 7: Thermal/stiffness loop of the Twin Turret design.

Using rotary axes as the primary machine axes enables complex geometrical traverse grinding as well as tool normal grinding without the need for an additional rotary axis. Multiple tools can be mounted upon a common and highly rigid turret. The turret shafts are precision ground and the very large diameter (440 mm) bearing provides excellent averaging producing radial and axial spindle error motions of around 100 nm

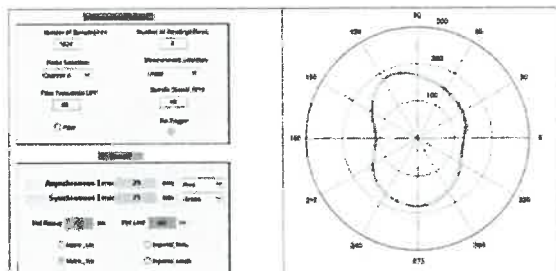


Figure 8: Axial turret error motion.

It is these rotary axes that provide the primary motion control for the machine, replacing conventional linear axes. They are extremely stiff, axisymmetrically.

Peripheral wheels are mounted with the grinding wheel aligned to the turret axis, thus wheel imbalance and process forces are directly resisted by the bearing stiffness, not the axis servo. For cup wheel grinding the grinding wheel to component interface is maintained at tool normal, thus any imbalance force induced axis oscillation produces only a 2nd order motion between wheel and component.

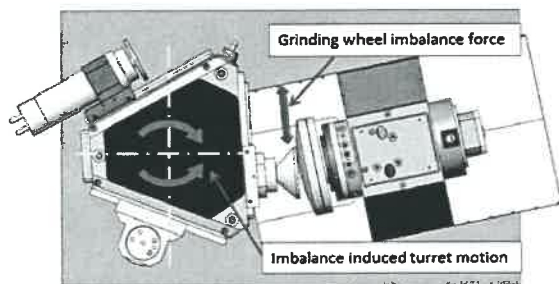


Figure 9: Wheel imbalance forces for cup wheel grinding.

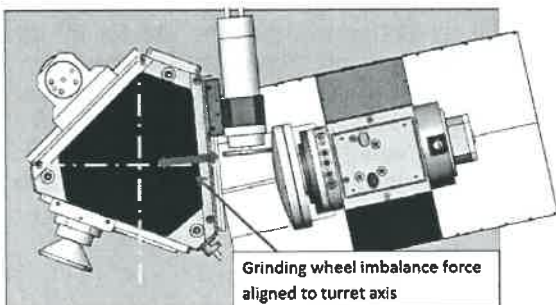


FIGURE 10: Wheel imbalance forces for peripheral wheel grinding.

GRINDING RESULTS

The grinding results presented here are for both cup and peripheral wheel grinding

Traverse ground aspheric surface

These results are for 'tool normal' grinding of an aspheric surface ground on a fused silica component. The traverse grind time using a metal bond wheel, Ø60mm, 6µm grit was 90 minutes. The form error of the ground surface is 280 nm P-V. This plot shows the result after a corrective run (using measurement data from Form Talysurf to provide the correction) and is thus effectively a measurement of the machine's repeatability, including thermal stability and process variability over 90 minutes.



FIGURE 11: Asphere grinding using cup wheel.

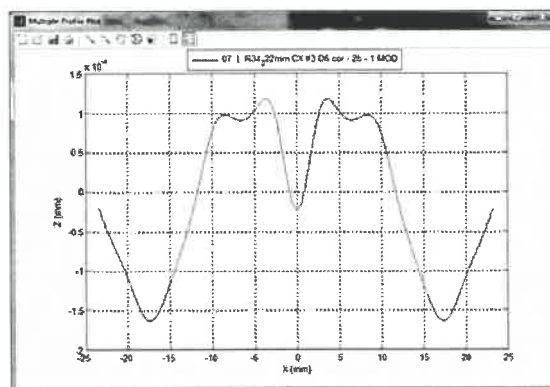


FIGURE 12: Ground asphere PV form error 280 nm PV (Form Talysurf, Zeeko Metrology Toolkit analysis).

Plunge ground spherical surfaces

The machine has been used to test the ability to manufacture spherical surfaces by plunge grinding (cup wheel mode shown in Fig 9). The images below show the results of a plunge grind using 14 μm diamond abrasive in a resin bond wheel.

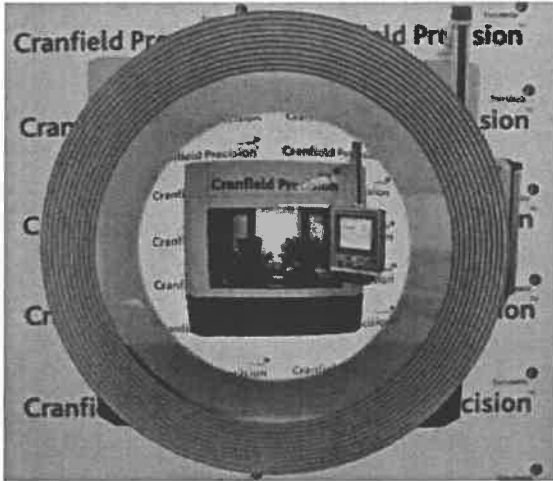


FIGURE 13: Image through a concave/convex lens plunge ground on TTOG machine.

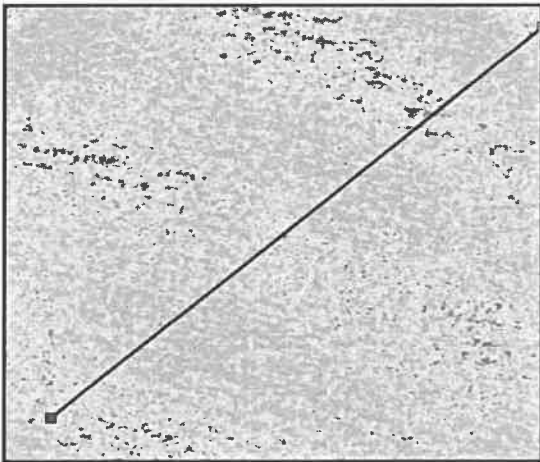


FIGURE 14: Surface texture measurement of plunge ground surface (shown in Fig 15)
 $R_a = 2.517 \text{ nm}$, $S_a = 3.191 \text{ nm}$

Off Axis ground freeform surface

By interpolating the motion between the two main turret axes, the linear infeed axis and the rotary workhead axis, it is possible to maintain tool normal grinding of complex freeform surfaces. The grinding tool traverse paths can be spiral or raster scan. The test part was a double saddle,

(concave and convex) inclined such that there is no line of symmetry on the surface.

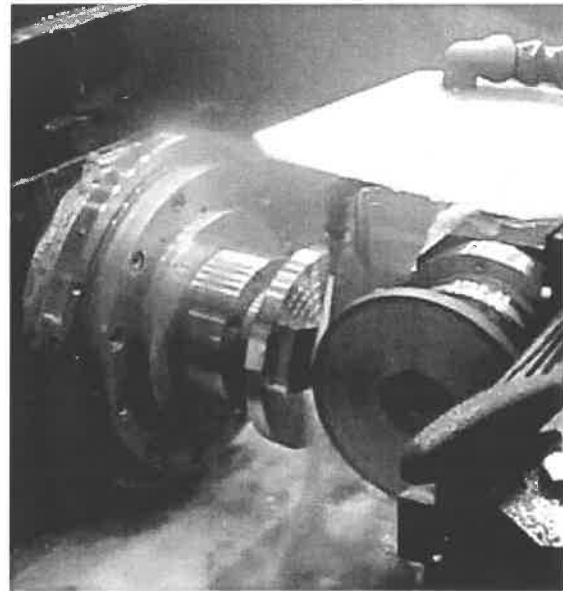


FIGURE 15: Ground off-axis freeform surface using a tool normal, interpolated raster scan motion path.

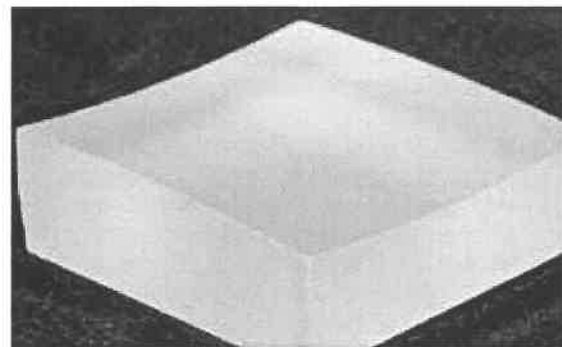


FIGURE 16: Rough ground surface (D42 metal bond wheel).

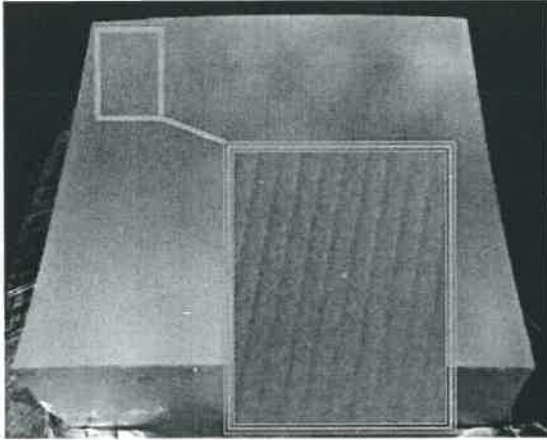


FIGURE 17: Rough ground surface shows raster scan path taken by grinding wheel.

Conclusions

A novel machining platform has designed and developed compatible with a wide range of machining processes. Grinding results demonstrate that the design objectives of ultra-high stiffness and thermal stability have been achieved.

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SOFT MOLD-BASED HOT EMBOSsing FOR HIGH-RESOLUTION PATTERNING ON NONPLANAR SURFACES

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In this paper, we present a new hot embossing process that enables high-resolution patterning of micro- and nano-structures on both flat and nonplanar substrates. In this process, a flexible elastomer stamp, i.e. PDMS, is used as a mold that performs hot embossing procedures on substrates of arbitrary curvatures. The new process was optimized through the development of an automated vacuum thermal imprinting system that allows precise control of all process parameters, e.g. temperature, pressure, and time. The new hot embossing process can be used to produce high quality 3-dimensional structures of various geometries. As a demonstration, we fabricated (1) hexagonal micro-lens arrays on a convex PMMA substrate and (2) 1.6 μm pitch optical gratings on a concave PMMA substrate. Surface quality and optical properties of the fabricated components were characterized quantitatively, i.e. $\text{RMS} \sim \lambda/30$, and proved to be comparable with high cost conventional precision processes such as laser lithographic fabrication.

HARD MOLD VS. SOFT MOLD

Hot embossing was first proposed by Chou in 1995 for patterning high resolution structures on thermoplastic materials [1]. The process utilizes a hard mold containing nanoscale features. When in operation, the mold and the substrate are heated to the glass transition temperature (T_g) and brought into contact to pattern polymers, e.g. PMMA, PS, PC, and PE, generating nanoscale features of 25 nm with good throughput [1]. However, using a hard mold may have the following drawbacks: (1) small features in the mold can be easily polluted and difficult to clean due to polymer adhesion, and (2) pressure distribution during molding is non-uniform, which limits the accuracy and area of replication and may even damage the mold. (Substrates are never flat at nanoscale.)

Compared with the conventional hot embossing process, a soft elastomer mold has the following

advantages: (1) elastomeric molds are low cost and can be easily replicated from a master; (2) the flexible mold enables conformal contact and uniform pressure distribution during molding; (3) elastomer molds, e.g. polydimethylsiloxane (PDMS), are chemically inert and anti-adhesive to polymers and can be easily demolded without cracking when compared with the hard mold.

SOFT MOLD PREPARATION

PDMS was chosen as the material for the soft mold in our new process because PDMS is thermally stable up to 200°C with low surface energy. Figure 1 illustrates the fabrication process of a PDMS mold. First, a master with hexagonal micro-lens array patterns was fabricated by a photolithography process and subsequently followed by a thermal reflow process. Then, a PDMS precursor mixture (10:1) was spin-cast on the master. After curing, the PDMS forms a soft mold of desired thickness.

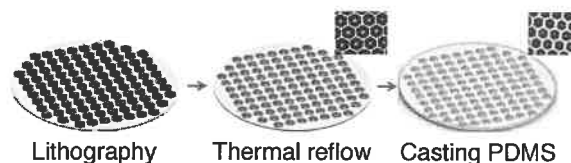


FIGURE 1. Illustration and images of the preparation of the micro-lens array master and PDMS mold

PATTERNING ON CURVED SUBSTRATES

Patterning micro- and nano-structures on nonplanar substrates has been technically challenging and prohibitively expensive via conventional processes, e.g. direct laser writing, where a substrate is fixed to a multi-axis positioner. Accordingly, new low-cost precision fabrication technologies can generate significant impact and enable a wide range of applications. A PDMS-based soft-mold is both flexible and highly stretchable. Thus, with a properly designed machine, it has the potential to pattern micro-/ nano-structures on nonplanar substrates.

In this work, we use the soft-mold based embossing approach to fabricate two types of optical components on curved PMMA substrates: (1) micro-lens array and (2) blazed gratings.

Precise patterning of micro-lens arrays on a convex substrate can form artificial compound eyes, i.e. a kind of micro-optics structure inspired by insect eyes, where thousands of micro-lenses are patterned on a convex surface with compact arrangement, pointing to slightly different directions. This device offers unique optical characteristics including ultra-wide field of view and high sensitivity [2]. The first low-cost compound eye structure was fabricated by Lee in 2006 using the polymer replication process; compound eyes were formed by filling the UV-curable resins to a deformed PDMS membrane with micro-lens patterns [3]. Another approach is to apply the thermal extrusion process, where a flat PMMA substrate containing micro-lens array is bent and extruded to form the compound eye structure [4]. However, none of these low-cost methods can produce uniform high-quality micro-lens arrays over a large area on substrates of arbitrary curvatures.

Concave gratings are blazed gratings patterned on a concave lens. A concave grating allows a grating to serve both as a dispersive element and a focusing element, integrating two optical components into a single, compact one. It is also possible to design aberration-corrected concave gratings which enable compact optical configurations to be developed for imaging and spectroscopic applications [5]. Conventionally, concave gratings are fabricated by mechanical ruling [6] or laser direct writing [7]. These techniques are slow, expensive, and cannot produce gratings of different blaze angles.

VACUUM THERMAL IMPRINTER DESIGN

To scale up and execute the new hot embossing process with precision, a vacuum thermal imprinting system was conceptualized and developed based on the vacuum imprinting system reported in [8]. Figure 2 shows the schematics of the imprinting system, where the PDMS mold is installed in the middle of the chamber, separating the room into two independent chambers (A and B). The substrate is placed in the bottom chamber. Independent air valves and pressure sensors are used to control the pressure difference between the top (A) and bottom (B) chambers. An infrared lamp

is installed in chamber A to provide heat for hot embossing. In addition, a load cell is integrated with the substrate holder in chamber B to monitor the printing force in real time. A thermocouple is installed in chamber B to measure and control the temperature.

Figure 3 illustrates the thermal imprinting procedure. Step 1: Both chambers are vacuumed and the IR lamp heats up the substrate to the molding temperature; step 2: Chamber A is pressurized to deform the flexible mold to perform a hot embossing process; step 3: The substrate is cooled down to the demolding temperature with controlled imprinting force; step 4: Demolding is performed by pressurizing chamber B.

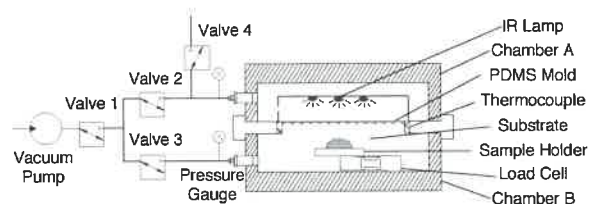


FIGURE 2. Schematics of the vacuum thermal imprinting machine

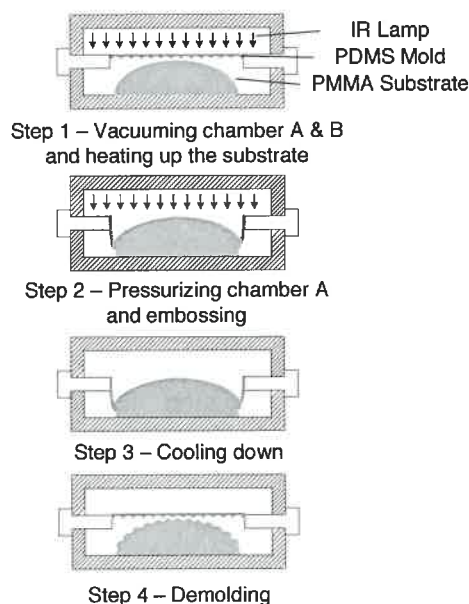


FIGURE 3. Illustration of the hot embossing process with a soft and flexible mold

The vacuum thermal imprinter presents a few distinct advantages for the soft-mold hot embossing process: (1) the vacuum process provides a particle-free environment and ensures air will not be trapped in small pockets of the mold during the embossing process, (2)

the controlled top chamber pressure ensures uniform embossing force and minimizes minor defects on the mold, (3) the molding parameters including embossing temperature, force and time can be precisely controlled, (4) precision prints can be performed on both planar and curved substrates.

PROCESS OPTIMIZATION

During the embossing process, undesired properties are mainly attributed to the uncontrolled flow behavior of the polymer (PMMA). A set of experiments was devised to study how various processing parameters, including temperature, pressure and hold time, can influence the forming of micro-/nano-structures. In these experiments, the embossing temperature varied from 120 °C to 260 °C, the imprint pressure varied from 0.5 psi to 4 psi, and the hold time varied from 30 to 240 seconds.

From the experiments, we learned that to ensure faithful replication of small features (<5 micron), especially high aspect ratio structures, the optimal embossing temperature is 180 °C, approximately 75 °C above the T_g of PMMA. The experimental results also indicate that the proper embossing pressure is between 2 and 3 psi. Low embossing pressure often leads to incomplete mold-filling as sufficient forces are needed to overcome the viscous force between the mold and PMMA. When the pressure is too high (> 3 psi), the soft mold can be distorted or collapsed, resulting in inaccurate pattern dimension. The optimal hold time is around 120 - 180 seconds, which ensures the complete filtration of the polymer in the soft mold. Note that excessive amount of hold time usually results in cracks in PMMA substrates due to internal stress. The results of the embossing parameter experiments are summarized in Figure 4.

FABRICATION AND CHARACTERIZATION

We first fabricated the hexagonal micro-lens array on a convex PMMA substrate. Figure 5 presents the experimental data for the thermal embossing process illustrated in Figure 3: Both chambers were first pumped down to 4 psi as the temperature was raised and maintained at 180 °C. During the embossing stage, the pressure in chamber A was increased and decreased in a cyclic fashion to remove the trapped air and improve overall embossing quality [8]; during the cooling stage, the temperature was lowered to 80 °C.

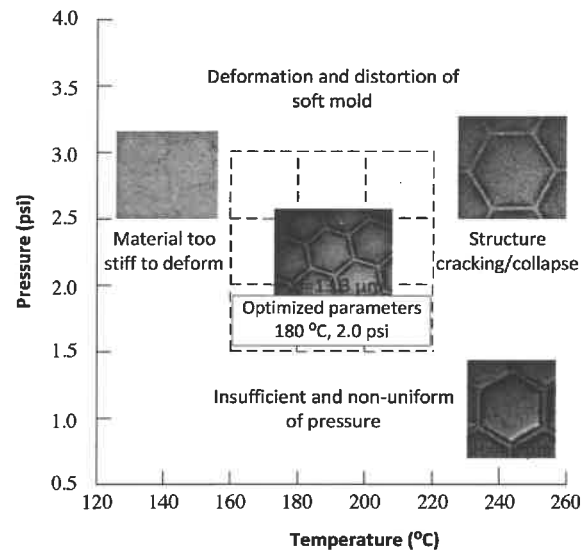


FIGURE 4. Embossing process optimization

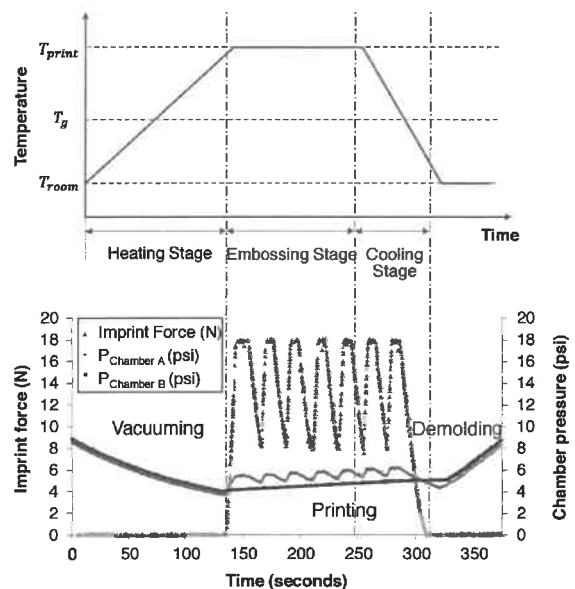


FIGURE 5. Temperature, force, and pressure readings from chamber A and B during the hot embossing process

Figure 6 shows an optical image of the hot-embossed micro-lens array on a convex PMMA substrate, which forms an artificial compound eye with no visible defects. An SEM image of a mantis's eye is shown next to the artificial compound eye as a comparison.

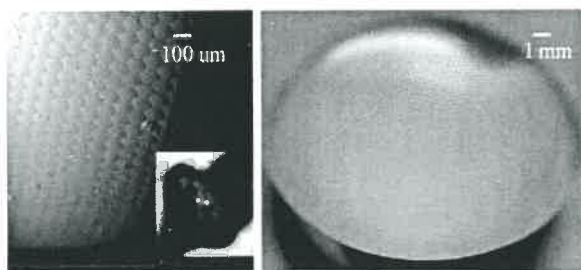


FIGURE 6. An SEM image of a mantis's compound eye (left) and an optical image of the artificial compound eye (right)

To examine the fabricated compound eye, a surface profiler (Tencor Instruments, Alpha-Step 500) was used to characterize the lens surface. Figure 7 shows the profiles of a micro-lens array pattern on the convex substrate and a randomly selected micro-lens. The measured diameter (D) of the micro-lens is $187 \mu\text{m}$ and the sag height (h) at the lens vertex is $13.29 \mu\text{m}$. (Micro-lens on the master: $D=185 \mu\text{m}$, $h=15 \mu\text{m}$)

We also measure the surface roughness using a Michelson interferometer ($\mu\text{Phase 2}^{\circledR}$ from FISBA OPTIK). Figure 8 shows the roughness analysis of a micro-lens from the fabricated compound eye. The observed deviation of the root-mean-square surface roughness (RMS) of the micro-lenses is 19.06 nm , which is less than $\lambda/30$. The results show the hot-embossed optical components possess outstanding optical quality.

The optical properties of the micro-lenses are determined by the radius of curvature (R), the focal length (f), the numerical aperture (NA) and the acceptance angle ($\Delta\phi$), which can be calculated using the following equations based on theories of geometric optics [9]:

$$R = \frac{D^2 + 4h^2}{8h}, \quad f = \frac{R}{n-1}, \quad NA = \frac{D}{2f}, \quad \Delta\phi \approx \frac{D}{f}$$

where the refractive index n of PMMA is 1.49. The calculated radius of curvature, the focal length, the numerical aperture and the acceptance angle of the micro-lens are $335.55 \mu\text{m}$, $684.79 \mu\text{m}$, 0.14 and 34.06° respectively.

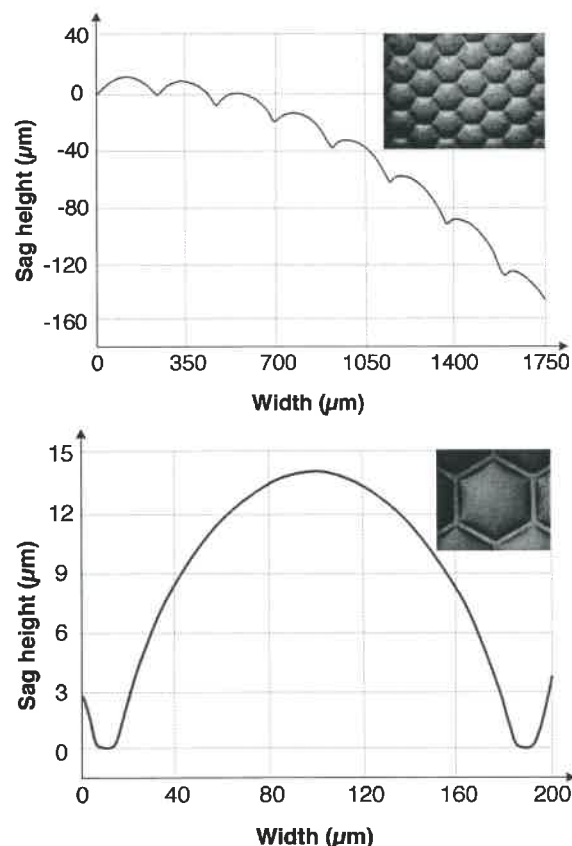


FIGURE 7. Cross-section profile of the micro-lens array (top) and a randomly selected micro-lens (bottom)

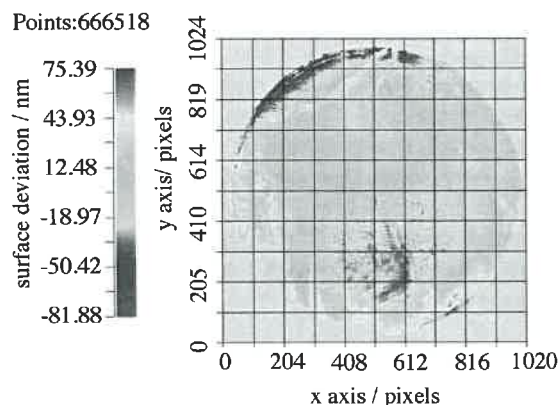


FIGURE 8. Geometrical testing for micro-lens in a Michelson interferometer: Deviation of a micro-lens from the measured specimens

In the second experiment, we fabricated a $1.6 \mu\text{m}$ pitch blazed grating on a precision plano-concave PMMA lens (Diameter: 25.4 mm ; focal length: 50 mm ; center/edge thickness: $3.5 \text{ mm}/6.8 \text{ mm}$). Figure 9 shows an image of the PDMS membrane containing a negative replica

of the blazed grating, where the blaze angle is $8^{\circ}38'$. Figure 10 shows an optical image as well as AFM characterization results of the concave grating fabricated by the thermal imprinter. The AFM results suggested grating patterns have been faithfully transferred to the concave lens and the soft-mold hot-embossing method has nanoscale resolution.



FIGURE 9. Soft PDMS mold with grating patterns replicated from a quartz grating master

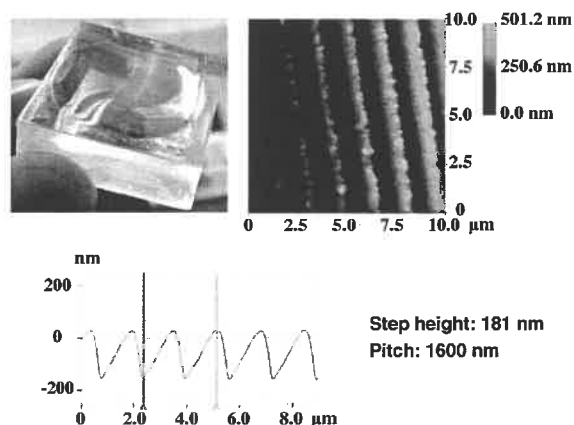


FIGURE 10. Optical Image and AFM characterization results of the concave grating

CONCLUSION

We have developed a new hot embossing process using soft PDMS molds that enable batch fabrication of high aspect-ratio precision micro- and nano-structures on nonplanar substrates at low cost. A vacuum thermal imprinting machine was constructed to precisely control the operating parameters. Experiments were devised to obtain the optimal embossing parameters (temperature: 180°C , pressure: 2 psi, hold time: 120 seconds). For demonstration, we successfully fabricated and characterized (1) micro-lens arrays on a convex PMMA substrate

which forms an artificial compound eye and (2) 1600 nm pitch blazed grating on a 1" PMMA plano-concave lens which forms a concave grating.

ACKNOWLEDGEMENT

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DESIGN AND DEVELOPMENT OF HIGH-FREQUENCY ACTUATOR FOR NANOCOINING

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INTRODUCTION

Structured surfaces found in nature have been the subject of research efforts to develop advanced surface coatings that can exhibit antireflective (AR), superhydrophobic or superhydrophilic properties. AR surfaces, which mimic the sub-micrometer surface features found on a moth's eye, are periodic structures that create a gradual index of refraction profile which increase the transmittance of a surface [1]. This technology is of interest for increasing the performance of devices such as optical lenses, solar cells and photosensitive detectors [2]. Existing techniques used to create sub-micrometer AR features, including lithographic techniques, etching, and bio-templating, are often costly and time-consuming on a large scale [3-5].

Nanocoining is a mechanical approach to rapidly create surface structures using a small diamond die ($20 \times 20 \mu\text{m}$) containing an array of 1600 nanofeatures machined with a focused ion beam (FIB). This die is pressed into a diamond-turned surface, leaving a field of sub-micrometer features (FIGURE 1).



FIGURE 1. A prefabricated diamond die is pressed into a diamond turned surface to create desired features.

To create a continuous area of features, the individual die indents must be tiled together. One way this is done is by indenting the surface of a rotating drum on a diamond turning machine (DTM) as shown in FIGURE 2(a). The die must be held by an actuator that pushes it into the surface to replicate the die features while moving tangentially at a speed that matches the surface of the drum during contact to avoid smearing of the individual features. The drum surface speed is controlled by the spindle motor

rpm and the radius of the drum. In addition to matching the drum speed during impact, the die must land one die width ahead of its position on the last cycle. This defines an "upfeed" distance that sets the relationship between the frequency of the die motion, its size and the surface speed of the drum. The "crossfeed" motion of the linear axis holding the die must advance along the drum at one die-width per revolution. Once the drum is uniformly covered with the desired structures, it is used as a mold in a roll-to-roll process to rapidly generate features on a pliable medium (FIGURE 2(b)).

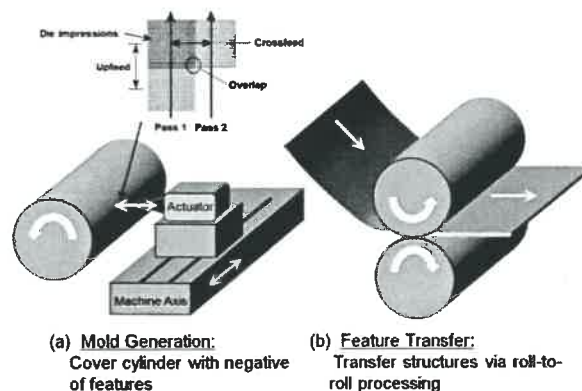


FIGURE 2. Nanocoining of sub-micrometer surface features on a cylindrical workpiece (a) and roll-to-roll replication using indented roller.

Since the die is small ($20 \times 20 \mu\text{m}$), indents must occur quickly (40 kHz) to cover 1 m^2 in reasonable amounts of time (14-18 hours). Generating tiled arrays of indents on the mold at 40 kHz requires the surface to move at a velocity such that it travels the length of the die ($20 \mu\text{m}$) during every period of the actuator oscillation ($25 \mu\text{sec}$). The work surface is moved a constant rate based on the actuator frequency and die size and the result will be a succession of indents precisely following the previously formed features. When indenting into a translating workpiece, the die must match the tangential velocity of the drum while the die is in contact with the surface. If this condition is not

achieved, the shape of the die features will be distorted or smeared.

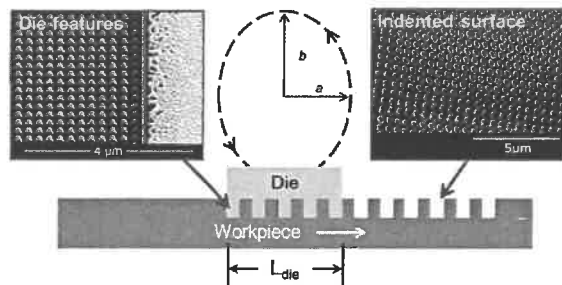


FIGURE 3. Vertical and horizontal dimensions of elliptical indenting path.

To minimize the distortion caused by a velocity mismatch, the die must be moved in a 2D path that allows the die to be pressed normal in to the surface while simultaneously matching the tangential speed of the workpiece. This goal can be achieved by moving the die in an elliptical path as shown in FIGURE 3. If the workpiece is moving at a constant speed, the die actuator motion can be designed to match that speed at the bottom of the elliptical path in FIGURE 3. However, there has to be some small indentation distance so the die and workpiece will contact over some finite time. This means that an elliptical path cannot achieve a pure indentation motion. However, if the ellipse is taller than it is wide and the indentation distance is on the order of 100 nm, the smearing of individual die features will be less than 10 nm.

ACTUATOR DESIGN

To achieve the desired elliptical motion at 40 kHz, a combined-mode resonant actuator has been developed. The concept is to design the geometry of a free-free beam-type actuator such that it is resonant in two orthogonal modes at the same frequency, resulting in elliptical vibration (FIGURE 4). This concept has been demonstrated in devices developed for machining processes such as elliptical vibration-assisted machining (EVAM) and ultrasonic welding that utilize resonant, 2-D vibrational motion [6-8].

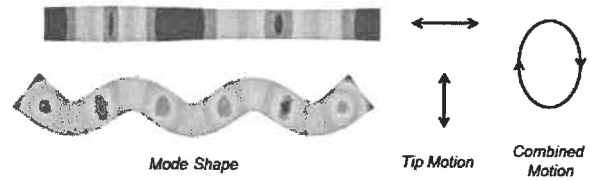


FIGURE 4. Combining longitudinal and bending motion to produce elliptical vibration path.

Geometry

The overall dimensions of the actuator are determined based on analytical solutions for a free-free uniform beam and are a function of the desired vibration modes and operating frequency. To increase the amplitude of the tool tip at resonance as compared to a uniform beam design, the geometry of the beam is reshaped to act as an ultrasonic concentrator. Ultrasonic concentrators reduce the cross-sectional area of the actuator and act as a strain amplifier. For simplicity, a stepped horn design was utilized to maximize vibrational amplitude in the longitudinal and transverse directions. The cross-sectional area was reduced at the tool tip as shown in FIGURE 5, where $h_1 > h_2 > h_3$ and $W_1 > W_2$.

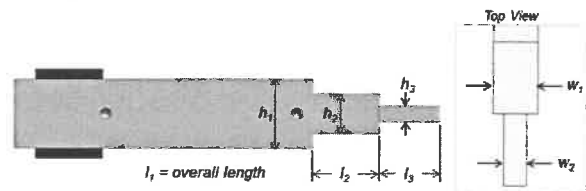


FIGURE 5. Beam geometry with concentrating horn design.

This geometry selection is also influenced by factors such as 1) the mounting points, and 2) the vibration mode frequencies. An iterative design method is used to optimize for these factors using finite element modeling.

Mounting Points

The actuator must be mounted in a manner that is axially stiff to prevent deflections during the indentation process. Additionally, the locations of the mounting points must not interfere with the resonant vibration of either the longitudinal or bending modes. Since the goal is to generate two directions of motion simultaneously, the selected mounting points must be located at vibration nodes that are mutual to the two desired mode shapes.

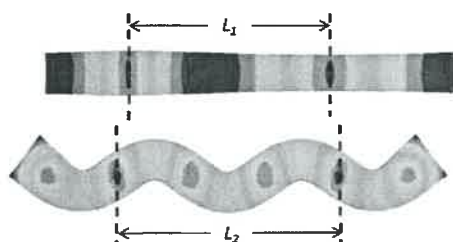


FIGURE 6. Displacement node locations for a uniform beam.

In FIGURE 6, the only two nodes for the longitudinal mode are separated by distance L_1 , whereas the two 'matching' nodes in the bending modes are separated by distance L_2 . Since L_1 does not equal L_2 , if the two mounting points were to be located at the nodes for the longitudinal modes, then the bending motion would be restricted by the mount contacts. Likewise, if the nodes were chosen for the bending mode (separated by L_2) the longitudinal mode would be restricted. To address this issue, the parameters h_2 , h_3 , l_2 , l_3 , and w_2 shown in FIGURE 5 are chosen iteratively using finite element modeling to align the desired nodes for mounting. Modal analyses can be performed on the unconstrained model to measure where the theoretical nodes are located. By adjusting the tip dimensions, two node locations on the bending and longitudinal modes can be made to converge such that they are suitable mounting points for both modes. An illustration of the final geometry with the nodes aligned is given in FIGURE 7.

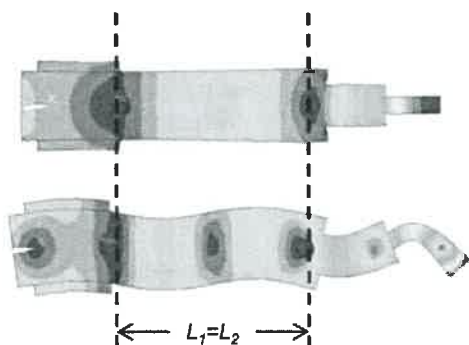


FIGURE 7. Actuator geometry modified for mutual mounting nodes.

Finite element simulations were used to investigate the sensitivity of the mount positions to the resonant frequency and amplitude of each operating mode by incrementally varying the mount position and simulating the actuator response. Shown in FIGURE 8, $L_{s,1}$ and $L_{s,2}$ are

the distances from the back of the actuator to the two mounting points.

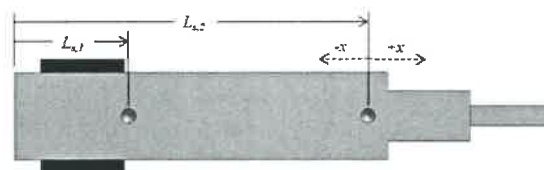


FIGURE 8. Actuator model for testing mount position sensitivity.

For this study, only the front node position ($L_{s,2}$) was varied where a positive increase in the x -direction means moving the mount closer to the tip of the actuator, while a negative increase in the x -direction means moving the mount closer to the rear of the actuator. The simulation results are presented in FIGURE 9, where the $x=0$ position is the original mounting position for $L_{s,2}$. The change in the resonant frequencies is given as a deviation from the original model results.

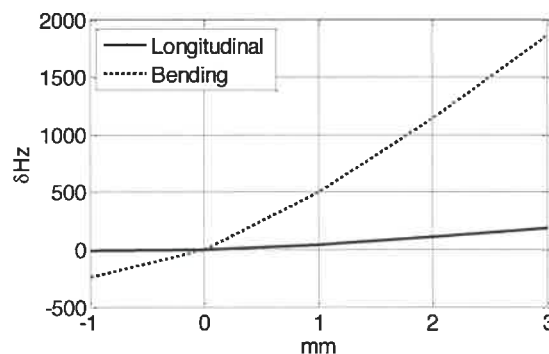


FIGURE 9. Simulation results of resonant frequency (a) for varying the position of $L_{s,2}$.

The results in FIGURE 9(a) show the deviation of the longitudinal and bending frequency from the original model, where $\delta Hz = 0$ and $x = 0$. The results show that the bending mode is quite sensitive relative to the longitudinal mode to changes in the mount position. Increasing x by 5 mm results in shift in the resonant frequencies between the two modes of almost 3 kHz.

These simulations demonstrate that the mount positions must be positioned correctly during the assembly of the actuator for the desired resonant behavior to occur. While the amplitude response shows little consequences for an error in the position, the mode frequencies appear to vary significantly with mount position. Based on the simulations, the mount locations must be

within 500 μm to keep the longitudinal and bending modes within 100 Hz of each other.

Tuning Techniques

Uncertainties such as stiffness of the epoxy connections, dynamic changes of that connection, node locations, and mount contact stiffness affect the resonant frequencies and are not easily captured in a the finite element model. Several methods have been developed for tuning these resonant frequencies such that both of the active vibration modes can be adjusted to occur at the same frequency despite model/fabrication differences.

When a difference in the two desired resonances is measured, a systematic method of moving the mode frequencies relative to each other has been developed. Methods such as selectively modifying the geometry of the actuator after assembly and strategically adding mass to the system can be used to tune the resonant frequencies. However, this can also change the location of the nodes which will result in a large decrease in amplitude.

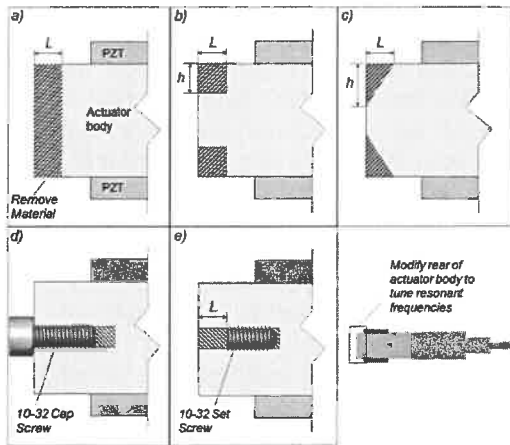


FIGURE 10. Configurations for altering the resonant behavior of the actuator.

FIGURE 10 illustrates various methods investigated to tune the resonant response of the system. FIGURE 10 (a), (b), and (c), involve removing material at the rear of the actuator to change the effective length and mass distribution of the system. FIGURE 10(d) and (e) illustrate methods of adding mass at specific locations.

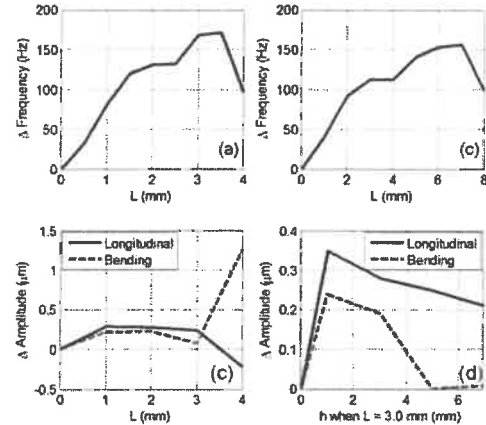


FIGURE 11. ANSYS results for frequency-tuning modifications where the bending mode is increased relative to the longitudinal mode.

FIGURE 11 illustrates these results for both changes in frequency and amplitude. The plots in FIGURE 11(a) and (b) are the relative change in resonant frequency between the longitudinal and bending modes from the changes shown in FIGURE 10(a) and (c), respectively, where $\Delta\text{Frequency}$ is the difference between the longitudinal and bending mode frequency. FIGURE 11(c) and (d) are the change in output amplitude of the actuator for the longitudinal and bending mode. Modeling the amplitude response is important to ensure that the tuning modifications do not adversely interfere with the behavior of the actuator. It should be noted that these methods increase the bending mode relative the longitudinal mode, so for instances where the initial response shows the longitudinal mode lower than the bending mode another strategy is needed.

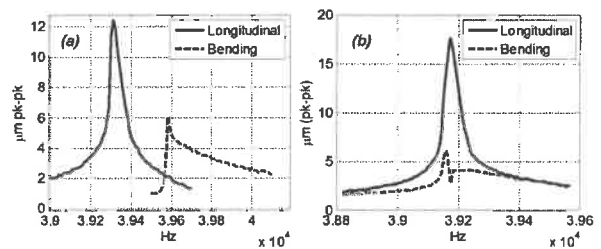


FIGURE 12. Untuned amplitude response (a) and tuned amplitude response (b).

The technique demonstrated in FIGURE 10(e) was used to modify the combined response of the two vibration modes in FIGURE 12. FIGURE 12(a) shows a difference of 270 Hz in the frequency of the two desired modes. This was changed by iteratively adding mass to lower

the transverse resonant mode and create the response shown in FIGURE 12(b). With the desired modes aligned, this configuration was used in indentation experiments.

ACTUATOR CONTROL

During extended operation, the resonant frequency of the system can drift due to piezo self-heating effects and other environmental influences. To maintain a consistent elliptical vibration path, a form of resonance-tracking control is needed to continuously update the drive frequency to the actuator. This is performed by measuring the phase between the current and voltage across the piezo inputs. Due to the interconnection of the piezoelectric actuators and beam, the electrical phase reflects the mechanical response of the beam. This concept is often referred to as minimum impedance tracking because mechanical resonance corresponds to minimum electrical impedance.

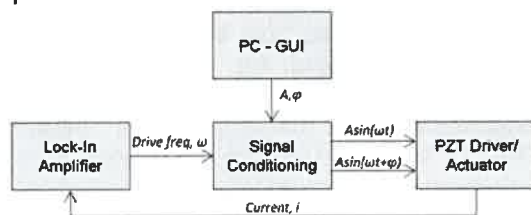


FIGURE 13. System to automatically modulate frequency ω to track actuator resonance.

A lock-in amplifier was used to automatically synchronize the drive signal frequency and the measured current signal. The output frequency is modulated to maintain a specified current-voltage phase that corresponds to system resonance. A GUI is used to condition the drive signal to power the two piezo elements out-of-phase (FIGURE 13).

EXPERIMENTAL RESULTS

Since the indents must be held consistently at a depth of about 500 nm and positioned to less than 1 μm relative to each other, a high-precision diamond turning machines are required to orchestrate the workpiece and actuator position. A cylindrical workpiece was mounted on the spindle of the DTM and cut with a single point diamond tool to produce an optical quality finish ($R_a < 10 \text{ nm}$). The actuator is mounted on the y-axis and aligned such that the die is square with the periphery of the workpiece. While the spindle is rotating the workpiece, the vibrating actuator is brought into contact with the workpiece using the x-axis, and

crossed in the z-direction to imprint the die pattern around the periphery of the workpiece (FIGURE 14).

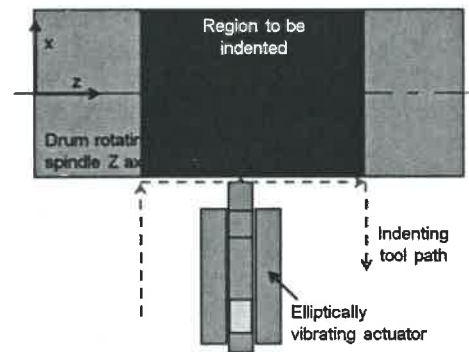


FIGURE 14. Top view of tool motion for ultrasonic indenting.

The nanocoin setup uses a Rank Pneumo Nanoform 600 with a Moore Nanotechnology Systems y-axis, resulting in a 3-axis (+ spindle) DTM shown. The workpiece is mounted on the spindle which moves in the x-direction while the actuator is mounted on the y-z axis.

Elliptical Path

A two channel MTI-2100 photonic sensor and a Tektronix MSO2014B oscilloscope are used to monitor the actuator motion in real-time. The actuator was driven with a two, 200V AC signals that are 100° degrees out-of-phase to excite both modes simultaneously. The actuator frequency was set to a frequency around 39,500 Hz; however, the actual drive frequency can be varied to change the relative motion between the two modes (i.e. emphasize one mode more than the other). The photonic sensors are positioned normal to the front and top face to capture both the longitudinal and transverse motion of the actuator. The elliptical motion used to perform the indents is shown in FIGURE 15.

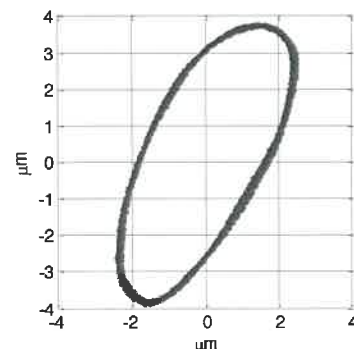


FIGURE 15. Measured elliptical indenting motion.

Indent Experiments

Indents are performed on a 360 brass workpiece using the 40 kHz actuator design. The indented workpiece was imaged using a scanning electron microscope (SEM) to examine the final indent results. Images of a selected region of indents are shown in FIGURE 16 for magnifications of 250x, 500x, 4000x, and 20000x.

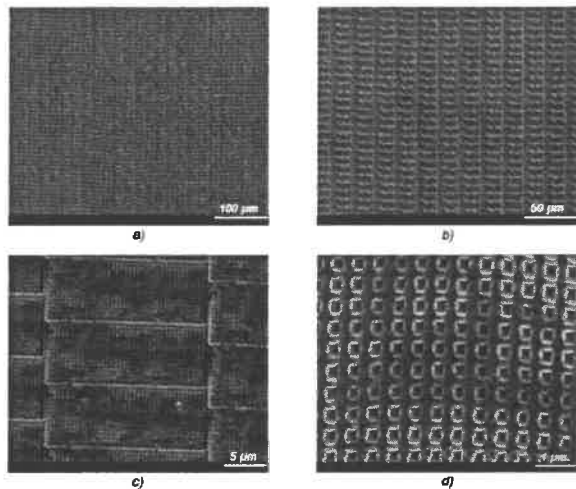


FIGURE 16. SEM images of indented brass workpiece at 250x (a), 500x (b), 4000x (c), and 20000x (d).

The indents in FIGURE 16 were generated with the worksurface moving from bottom to top, and the actuator crossfeeding from left to right. Each die impression appears rectangular (in spite of using a square die) due to the fact that overlapping was required to match the worksurface speed with the attainable transverse velocity of the actuator. As transverse amplitude of the actuator is increased, larger surface velocities can be used to match the speed.

The lower magnification images in FIGURE 16(a) and (b) demonstrate the ability to achieve uniform areas of indents using a resonant actuator and appropriate feedback controller. The higher magnification images in FIGURE 16(c) and (d) clearly show the impressions left by the square-post die. The indents appear well-formed with minimal distortion, suggesting that the worksurface speed matched the velocity of the die at the time of contact. The square borders around each indent are artifacts of the die, where a slight tilt in the orientation of the die relative to the workpiece causes one side of the die to indent slightly deeper. This alignment can

be fine-tuned in successive experiments by measuring the tilt of each indent and adjusting the actuator attitude appropriately.

CONCLUSIONS

Generating structured surfaces using nanocoining requires an elliptically vibrating, 40 kHz resonant actuator. This motion is achieved by designing the actuator such that two orthogonal vibration modes occur at the same frequency. However, due to model inadequacies, a systematic method of tuning the actuator is required to optimize performance. A control system was developed to automatically track resonance and maintain desired actuator behavior. The effectiveness of the system is demonstrated by rapidly generating sub-wavelength surface structures over a large area.

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Synchronization System of Galvanometer Scanner and 2-axis Linear Stage for Laser Routing in Wide Area FPCB coverlay

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ABSTRACT

In this current research, the two-axis galvanometer laser scanner and the two-axis linear stage system are synchronized for wide area laser machining with high machining speed. This synchronization system is used as the laser routing machine at consistent machining speed, 170mm/s with a 350×350 mm² area, which area is much larger than the galvanometer scanner working field. The coverlay of FPCB (Flexible PCB) is routed and accuracy is measured as about 5μm, which is similar error regime of a galvanometer scanner itself.

INTRODUCTION

Recently, laser machining is being employed increasingly in many diverse micro fabrication areas such as biomedicine, automotive manufacture, display devices, and semiconductors [1]. A galvanometer scanner delivers laser beam at certain position by rotating two mirrors using galvanometer motors with high speed. However, it has limitation of working area maintaining same optical accuracy. If a focal length which is distance from a f-theta lens to a sample is longer, working area may be larger but optical accuracy becomes worse. A linear stage system can guarantee high positional accuracy with wide area but it has limitation of fabrication speed. To combine two individual systems, step-and-scanning method is popularly applied. During a galvanometer scanner fabricating a sample, a linear stage system stops. After finishing fabrication of certain working area of a galvanometer scanner, a linear stage moves next step and a galvanometer scanner fabricates again. This method always meets discontinuities problems at intersection of scanning area and obviously slow machining speed [2].

To overcome this problem, we dynamically combine and synchronize two systems. By this

method, we routes continuously without any stopping motion and routing area is extended to two-axis linear stages area without drawback of machining speed.

METHODOLOGY

The synchronization system is built with the 3-axis linear stage, the 2-axis galvanometer scanner, the 355nm ns second laser source, vision system, and controllers which includes the motion controller, the scanner control board as shown Fig.1. Position and speed of the linear stage is fed to the scanner control board by the 2-axis encoder signals. In the scanner control board, scanner movement is calculated by comparison original CAD data and stage movement. The micro vectors of the galvanometer scanner with 10μs intervals always compensate micro motion of the linear stages. In general, stage motion is slower than scanner motion due to its heavy weigh.

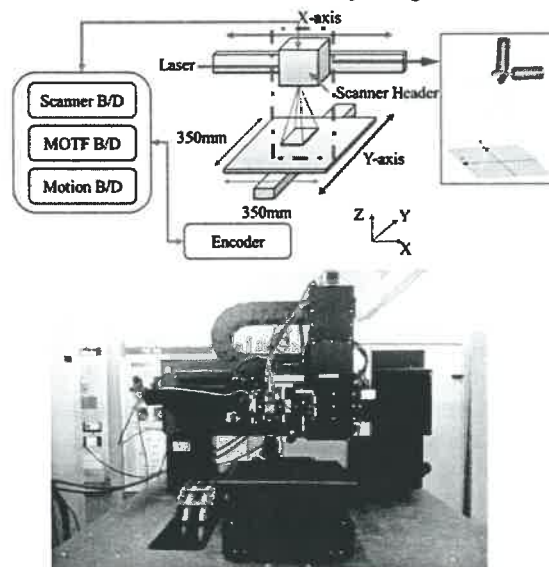


FIGURE 1. Schematic sketch & experimental built up of the laser synchronization system.

RESULTS AND DISCUSSION

Before operating the laser routing machine, macro motion of the linear stage and micro motion of the galvanometer scanner is separated. As shown in Fig.2, FPCB (Flexible PCB) coverlay of the smart phone camera module is planned to be routed. Size of coverlay sheet is $350 \times 350 \text{ mm}^2$, which is much larger than working area of the galvanometer scanner, $50 \times 50 \text{ mm}^2$. The stage motion is selected to be covered whole coverlay sheet and maximum speed and acceleration capability is considered. Due to the low inertia of the linear stage, motion and speed change should be minimized. Even though working area of the galvanometer scanner is narrow, the galvanometer scanner has high inertia which is over 10g in general. Path of it is then suitable for high frequency motion and quick change of speed. Most important thing for path separation, summation of the linear stage speed and the galvanometer scanner speed is constant to maintain constant routing speed. High routing speed can cause poor cutting edge or low routing speed can make cutting edge burn.

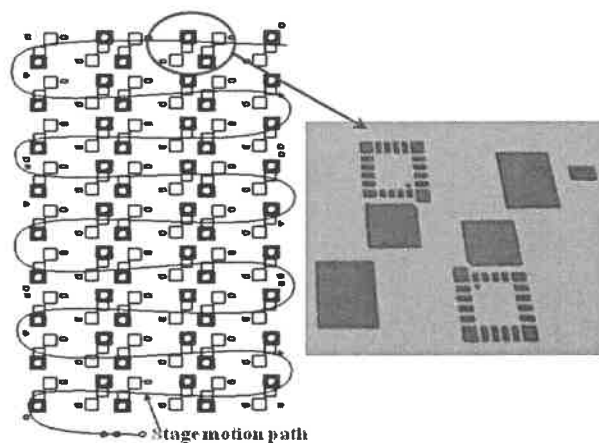


FIGURE 2. Routing path of synchronization system and photo of routed coverlay sample.

In Fig. 3, some routed FPCB coverlay samples are selected and measured side length and diameter. Most of samples are within $5 \mu\text{m}$ error range. Maximum error is measured as $5.61 \mu\text{m}$ in sample 3. When the sample is routed with only the galvanometer scanner, the error regime is measured in similar range.

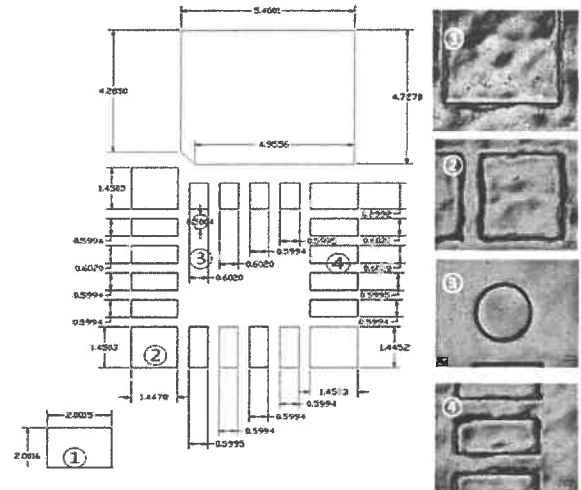


FIGURE 3. Photo of some routed samples and measurement data

CONCLUSION

The two-axis galvanometer laser scanner and the two-axis linear stage system were synchronized for wide area laser machining with high speed. Macro motion of the stage movement was planned with its maximum speed and inertia. In other hands, motion of the galvanometer scanner was planned with a high frequency in small working area. The synchronized laser machine was applied to route the FPCB coverlay sample with a 170 mm/s speed. The accuracy was measured in about $5 \mu\text{m}$, which was similar error regime of a galvanometer scanner itself.

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Fabrication of a Flexure-based Linear Translation Stage with Integrated Optical Position System

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For certain applications, such as a Fourier-transform spectroscopy, the position of a linear translation stage (shuttle) is precisely monitored using optical means. Typically, this is achieved by attaching an optical interferometer to the moving shuttle and to the companion fixed reference frame. This multi-subsystem approach is susceptible to both short and long-term misalignment when subjected to shocks, vibrations, or thermal effects.

Fabricating the moving platform and the rest frame from a single monolithic slab, and integrating the optical interferometer into this system, would alleviate these shortcomings. This would, however, require that the monitoring optical system be fabricated (at least in part) directly onto the translation stage or inside the translation stage. The later embodiment would further require that the stage be made out of a material that is transparent, but it would provide the additional advantage of an encapsulated optical system protected from the elements. This is the approach we decided to pursue.

Our monolithic linear translation stage with an integrated subsurface optical position system (see Fig. 1) is made of amorphous fused silica glass, also known as fused quartz.

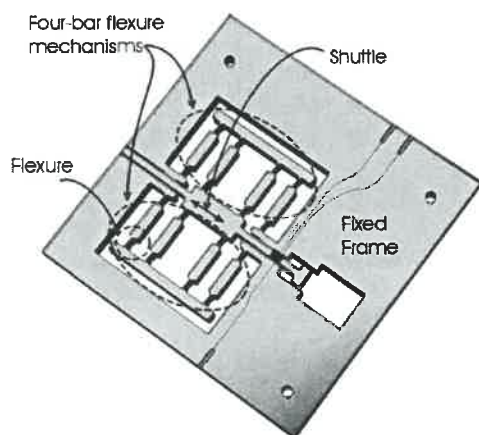


FIGURE 1. Model of our linear translation stage with integrated optical position sensor

Fused silica is a high-quality optical material, transparent from about 170 nm to above 3 μm . While commonly used in optical and chemical applications (where it is a material of choice due to its general chemical inertness), fused silica is rarely, if ever, used for its mechanical characteristics.

Fused Silica Micromachining

The fabrication of our linear translation stage, with its many flexure elements and its integrated optical position sensor, was made possible only by recent developments in machining fused silica, with the use of femtosecond lasers.

Our translation stage demonstrator is made out of a single piece of amorphous fused silica glass. The body of the demonstrator is precision-machined using a two-step process, which combines femtosecond laser illumination and chemical etching (see Fig. 2).

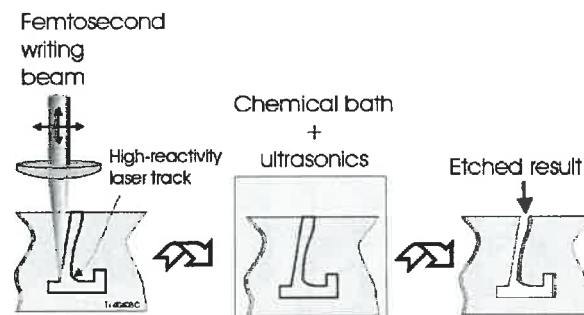


FIGURE 2. Two-step glass precision machining process

This process, first demonstrated by Marcinkevicius et al. [1], relies on the fact that the chemical resistance of fused silica to selected etchants can be greatly altered, and made to be highly non-isotropic, using femtosecond laser pre-irradiation. We refer to this process as *femtoEtch* [2]. The etching selectivity ratio between exposed and non-expose approaches 100:1 when using HF as an etchant, and it can exceed 1000:1 when using KOH, as recently demonstrated by Hermans [3].

Using the *femtoEtch* process, one can create almost arbitrary-shaped 3D volumes in fused silica (See Fig. 3).

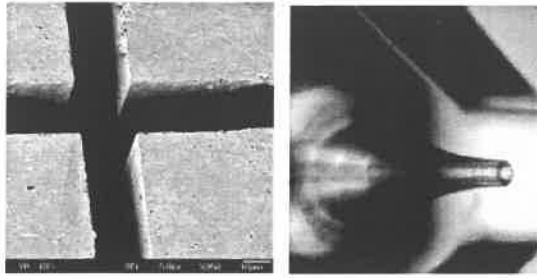


FIGURE 3. Two fused silica devices fabricated using the *femtoEtch* process. (Left) SEM image of a structure exemplifying edge quality. (Right) Micro-photograph of a shaped nozzle (ID 50- μm) fabricated within a microfluidic channel.

Although fused silica glass is intrinsically a strong material, it has not been previously used to fabricate mechanical flexures. This is mainly due to the heretofore absence of a suitable machining technique capable of generating the desired geometric profile, while at the same time preventing the formation of surface defects. Tomozawa [4] has shown that fused silica surface flaws act as stress concentrators and cause the nucleation of cracks, which ultimately lead to failure. Tomozawa also showed that most glasses, including fused silica, are prone to static fatigue under constant loading, due to stress-corrosion effects, *i.e.*, spontaneous crack growth leading to a reduction of the fracture strength in the presence of moisture. This effect can be significantly reduced using chemical treatment that blunt crack tips [4].

We have shown that our *femtoEtch* process does not generate noticeable surface flaws. In fact, it removes or blunts surface flaws that are present in the glass slab prior to our manufacturing; these flaws are typically generated during dicing, lapping, and polishing operations, which are commonly applied to glass pre-forms. The absence of surface flaws allows for the use of fused silica as an elastic material. Bellouard [5] has shown that our process can be used to fabricate unusually strong flexures, as shown in Fig. 4. Flexural strengths as high as 2.7 GPa were measured, a value that is greater than the breaking strength values of many common engineering materials. As part of this study, it was shown that prolonged etching time clearly enhances the maximum strength of the flexure.

A second fabrication process is used to fabricate the waveguides that are part of the optical

position sensor. Davis et al. [6] demonstrated the use of femtosecond lasers to locally increase the index of refraction of fused silica. This process (*femtoWrite*) is closely related to our *femtoEtch* process. Most critically, it uses the same femtosecond laser source.

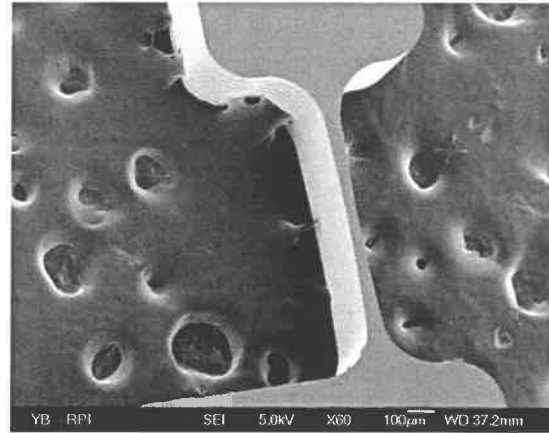


FIGURE 4. SEM image of a fused silica flexure fabricated with our two-step process (Photo courtesy of Dr. Y. Bellouard).

This is a key aspect of our fabrication approach: the various structural, mechanical, and optical elements are all introduced with the same direct-write laser workstation in a continuous manufacturing process. Specifically, there is no need for repositioning of the work piece. Consequently, the precision of this manufacturing process depends, to first order, only on the performance of the motorized stages used to move the specimen under the laser beam.

Microstage Design

Our microstage is composed of two main subsystems: (i) a flexure-based mechanism, which guides the motion of the platform along one axis, and (ii) a waveguide-based optical system that senses the ensuing displacement (see Fig. 5).

The mechanical design is a conventional double-compound rectilinear kinematic structure (see Fig. 6), sized to provide the desired 1-mm travel range without unduly stressing the flexing elements (see Table 1).

TABLE 1. Critical Mechanical Dimensions

Notch hinge thickness	50 μm
Notch hinge radius	7 mm
Distance from notch hinge center to next notch hinge center	4.7 mm
Fused silica plate thickness	1 mm
Young's modulus	75 GPa

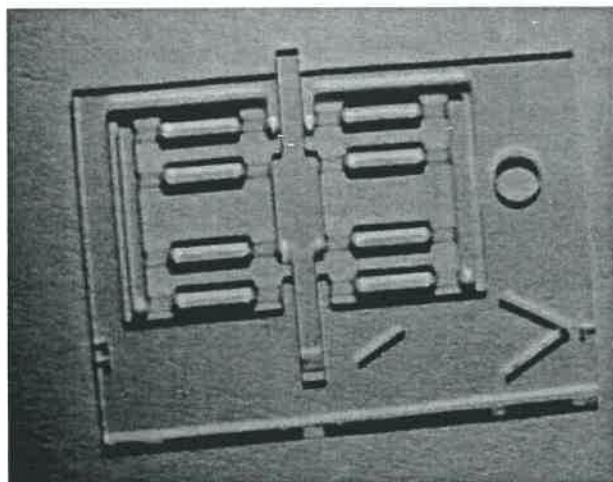


FIGURE 5. Demonstrator fabricated in a 1-mm thick fused silica substrate. A force applied to the shuttle tip induces a linear motion of the mobile platform

This design assumes small deformations, a pure elastic material behavior, and an elastic limit of 300 MPa; the device was over-etched to provide a high ultimate tensile strength. An analytical model predicts a force of 0.2N was required to reach the maximum excursion [7].

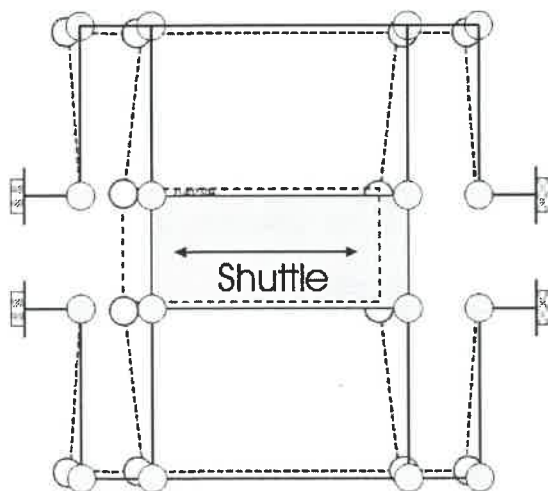


FIGURE 6. Double-compound design. The circle represents ideal mechanical joints with one degree of freedom (rotation in the plane).

We used signal intensity variation, induced by lateral misalignment between waveguides, as the basic principle to track the mobile platform displacement. The position monitoring subsystem is an integrand part of the fused silica platform. It consists of several optical waveguides: (i) an array of short optical waveguides, which are embedded in the moving platform, and (ii) longer waveguide segments embedded in the stationary frame, as illustrated

in Fig. 7. The waveguides are located mid-plane in the fused silica substrate. They have an $8\mu\text{m} \times 8\mu\text{m}$ square cross-section. The index refraction of the waveguide core is approximately 0.6 percent higher than that of the surrounding material. This index difference corresponds to that found in standard SMF-28 telecommunication. The waveguides are highly multimode at the operating wavelength of 670nm, but single mode at 1550-nm, albeit wavelength is not critical and selection of the source is dictated by other criteria, such as availability of reliable fiber-pigtailed optoelectronic components. The waveguide array consists of parallel waveguide segments, separated by 30 microns and spanning the 1-mm tail-end section of the movable platform. By design, one transmitting and one receiving waveguide, located in the stationary rest frame of the device, are in direct axial alignment with one of the array waveguides when the stage is unloaded and at rest, as illustrated in Figure 7. An input optical signal is coupled to the device using an optical fiber; fiber ports are machined in the rest frame. This signal travels successively along the input waveguide segment located in the rest platform, through a 20-micron free space gap (Fig. 8), through one of the waveguides found in the movable shuttle, through a second free space gap, and finally along the output waveguide segment, which is connected via an optical fiber to a photodetector.

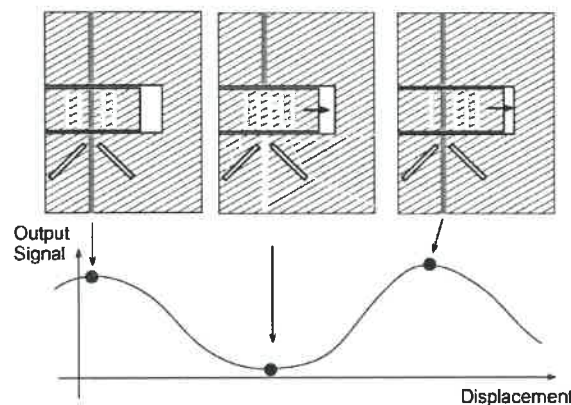


FIGURE 7. Operating principles of the waveguide-based position monitoring system. (Top) three geometries with changing relative displacement between the moving carrier and the rest frame. (Bottom) Corresponding optical signal.

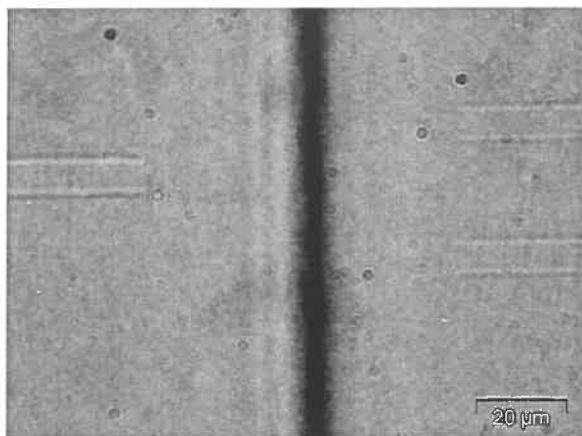


FIGURE 8. Close-up view of waveguide segments located near the junction between the rest platform (left) and the moving carrier (right)

The 30-micron spacing between waveguides in the movable array, create an uneven coverage of the range of motion (see Fig. 9). In more recent demonstrators, a second output waveguide, positioned in quadrature, was added to address this shortcoming. Various air gaps or slots, positioned in such a way to act as scattering blocks through total internal reflection, were also added to improve the optical signal-to-noise ratio. With this improved arrangement, position measurement resolution along the main axis was found to be better than 50 nm.

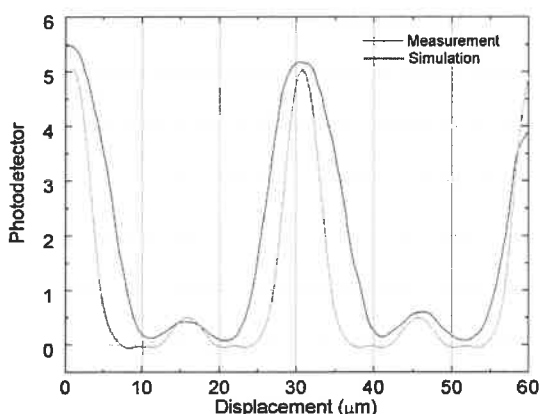


FIGURE 9. Experimental results (upper blue curve) compared with simulation results (lower green curve). With only one output waveguide, the displacement resolution is function of the displacement

Higher precision can be obtained by creating an optical micro cavity whose length is defined by the separation between the end tip of the moving platform and the stationary platform. A hybrid implementation, combining optical fibers with a glass micromechanical flexure-based structure, was recently demonstrated by

Cervantes. [8]. A fully integrated version of this approach can be implemented using integrated waveguides instead of surface-bond optical fibers.

In all of our demonstrators, the actuator was external to the device. Integrated electrostatic drives have been fabricated and tested independently. However, it was found that they could not provide the desired translation range. For many applications, an external actuator is acceptable as backlash and other assembly imperfections do not, to first order, critically affect the device operation as the position system is integral to the platform.

Sensitivity to thermal effects is generally not a significant issue either, as the coefficient of thermal expansion of fused silica ($3.5 \times 10^{-7} / ^\circ\text{C}$ @ 20 $^\circ\text{C}$) is one of the lowest of any solid.

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MECHANICAL DESIGN OF MULTIPLE FRESNEL ZONE PLATES PRECISION ALIGNMENT APPARATUS FOR HARD X-RAY FOCUSING IN THE TWENTY-NANOMETER SCALE

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INTRODUCTION

Fresnel-zone-plate-based optics is extensively applied for x-ray instruments [1]. At the Advanced Photon Source (APS) at Argonne National Laboratory (ANL), many synchrotron radiation beamlines are using Fresnel zone plates for hard x-ray focusing. However, the efficiency of Fresnel zone plates (FZPs) as focusing optics for x-rays depends on the height of the structures. In the hard x-ray regime, very high aspect ratios are required for maximum efficiency with focusing spot sizes of few tens of nanometers, which is required for future hard x-ray nanoprobe beamlines planned as part of the APS Upgrade project [2,3].

To overcome the limitations of today's fabrication techniques for high-efficiency hard x-ray FZPs, a new approach of stacking FZPs at larger distances (in an intermediate-field) was proposed by Vila-Comamala et al. in 2012 [4]. According to this new approach, stacking zone plates with large separation distance is possible by adjusting the diameter of the downstream FZP so that its focal length is equal to the focal length of the upstream FZP minus the distance between both FZPs. Thus, the focal spots of both FZPs overlay when the separation of both FZPs is matching the difference in focal lengths. However, besides designing and fabricating of high quality FZPs for intermediate-field stacking, there are many mechanical design challenges to transfer the theory to a practical instrument. First of all, a precision alignment apparatus for multiple FZPs handling and aligning must be designed to meet the following challenging design requirements:

- Each of the stacking FZPs needs to be manipulated in three dimensions with nanometer-scale resolution and several millimeters travel range.
- The relative three-dimensional stabilities between all of the stacking FZPs (especially in the x-ray beam transverse plane) are required to be kept within few nanometers

for more than eight hours, the duration of a typical hard x-ray nanoprobe operation.

- Compatible with the operation of multiple optics configuration for the APS future x-ray nanoprobe design.

To meet the demanding mechanical requirement for the precision alignment apparatus system for hard x-ray focusing in the twenty-nanometer scale, several prototypes have been designed and tested at the APS.

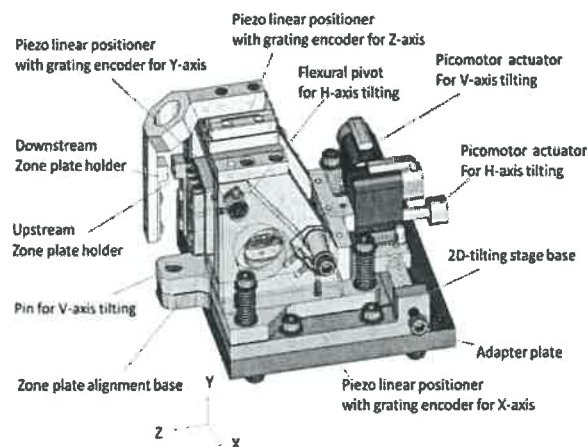


Figure 1. A 3-D model of a prototype of Z2-33 alignment apparatus for two FZPs stacking.

APPARATUS FOR TWO FZPS STACKING

Figure 1 shows a 3-D model of a prototype of alignment apparatus Z2-33 for two FZPs stacking [4]. It enabled the first experiment of stacking FZPs with adjusted diameter in the intermediate field, and the results prove the simulations by Vila-Comamala et al. [4]. It includes a pair of commercial Piezo-motor-driven linear stages (SmarAct™ SLC-1720S), which are mounted on the zone plate alignment base to provide 2-D alignment for the upstream zone plate in X-Y plane. The downstream zone plate holder is driven by a SmarAct™ SLC-1720S linear stage in Z-direction to adjust the

gap distance between the upstream and downstream zone plates. All of the three piezo linear positioners are mounted on a zone plate alignment base frame, which is a part of the carriage of a 2D-tilting stage. Driven by a picomotor™ actuator, the V-axis tilting stage rotates around a vertical pin, which is fixed on the base of the 2D-tilting stage and sliding fitted with the base of the H-axis stage. The H-axis stage tilts around a pair of flexural pivot as shown in figure 1.

APPARATUS FOR THREE FZPS STACKING

Figure 2 shows a 3-D model of a prototype of alignment apparatus for three FZPs stacking [5,6]. Its non-symmetric design is also compatible with mirror-based nanofocusing optics, such as Kirkpatrick-Baez (K-B) mirrors [7] for hard x-ray nanoprobe in switchable multi-optics operation modes. As shown in figure 2, the Z2-34 alignment apparatus has a non-symmetric invar base structure (1) and six commercial Piezo-motor-driven linear stages (SmarAct™ SLC-1720S). Three zone plates (2-4) are mounted on CVD-diamond holders (5-7). The CVD-diamond holder (5) for upstream zone plate (2) is driven by a stage (8) to adjust its position in Z direction with nanometer scale and stability. The second downstream zone plate (3) is driven by a pair of stages (9,10) to adjust its position in X and Y directions. The third downstream zone plate (4) is driven by a set of stages (11-13) to adjust its position in X, Y, and Z directions. Since the thermal expansion coefficient of CVD diamond is similar to the thermal expansion coefficient of invar, it basically ensures thermal stability of the apparatus. To further compensate the thermal deformation from the stages (8), (9,10), and (11-13), the materials of the linkage components (14-16) between the stages and CVD-diamond holders are carefully chosen. If it is necessary, two or three materials may be combined to compensate the stages thermal deformation precisely. Figure 3 shows a photograph of the Z2-34 alignment apparatus for three FZPs stacking.

APPARATUS FOR SIX FZPS STACKING

The Z2-37 precision alignment apparatus is designed for six zone plates intermediate-field stacking [6]. It is especially useful for hard x-ray focusing with x-ray energy 25 keV and above.

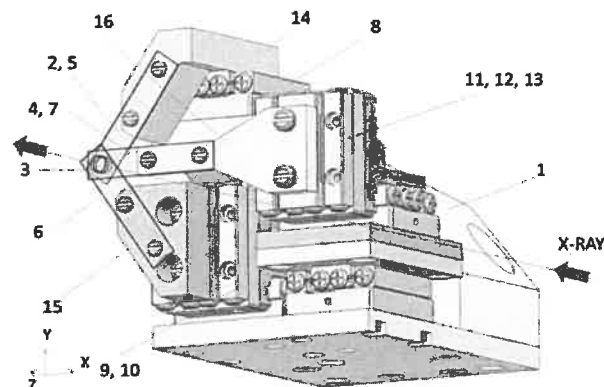


Figure 2. A 3-D model of the Z2-34 precision alignment apparatus for three zone plates intermediate-field stacking: (1) non-symmetric invar base structure; (2-4) zone plates; (5-7) CVD-diamond holders; (8) Z-stage; (9, 10) X-Y-stages; (11-13) X-Y-Z-stages; (14-16) linkage components.



Figure 3. A photograph of the Z2-34 alignment apparatus for three FZPs stacking test at the APS 2-ID-E hard x-ray experiment station.

As shown in figure 4, the Z2-37 alignment apparatus has a hexagon symmetric invar base structure (1) with an interface mounting plate (38) and eighteen commercial Piezo-motor-driven linear stages (such as SmarAct™ SLC-1720S, PI™ LPS-24, or Micronix™ PPS-20 stages). Six zone plates (2-7) are mounted on CVD-diamond holders (8-13). Each of the zone plate CVD-diamond holders is driven by a set of stages (14-16, 17-19, 20-22, 23-25, 26-28, and 29-31) to adjust its position in X, Y, and Z directions. Similar to the zone plate holders for Z2-34 alignment apparatus, the zone plate holders for Z2-37 alignment apparatus are made of CVD diamond material to ensure thermal stability of the apparatus. Materials of the linkage components (32-37) between the stages and CVD-diamond holders are also carefully chosen to further compensate the thermal deformation from the stage set. If it is necessary,

two or three types of material may be combined to compensate the stages thermal deformation precisely. Figure 5 shows a 3-D model of the Z2-3701 X-Y-Z stages sub-assembly module with CVD diamond holder and linkage component. Figure 6 shows a result of thermal structural stability study with finite element analysis (FEA) method for the zone plate thermal drifting in Y direction under a 1 °C temperature variation. Table 1 summarizes the results with different Y-stage positions. The results show that, under a 1 °C temperature variation, with SS304 linkage and AL6061 adapters, the zone plate center on the CVD diamond holder will have a less than 3 nm drift in a Y-stage travel range of -1 – 0 mm. Similar design with an octagon or decagon symmetric invar base structure and twenty-four or thirty commercial Piezo-motor-driven linear stages can perform eight or ten zone plates intermediate-field stacking for high energy x-ray focusing applications.

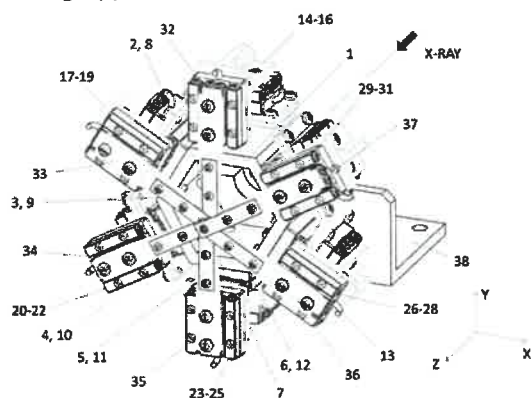


Figure 4. A 3-D model of the Z2-37 precision alignment apparatus for six zone plates intermediate-field stacking: (1) invar base structure; (2-7) zone plates; (8-13) CVD-diamond holders; (14-31) X-Y-Z stages; (32-37) linkage components; (38) an interface mounting plate.

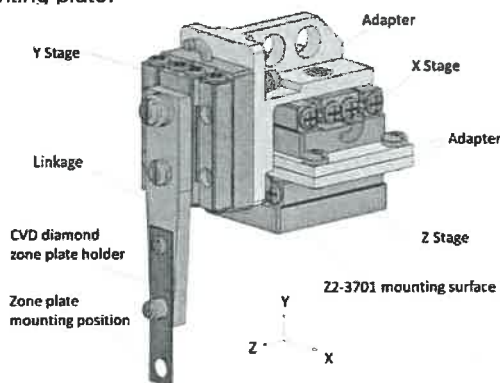


Figure 5. A 3-D model of the Z2-3701 X-Y-Z stages sub-assembly module with CVD diamond holder and linkage component.

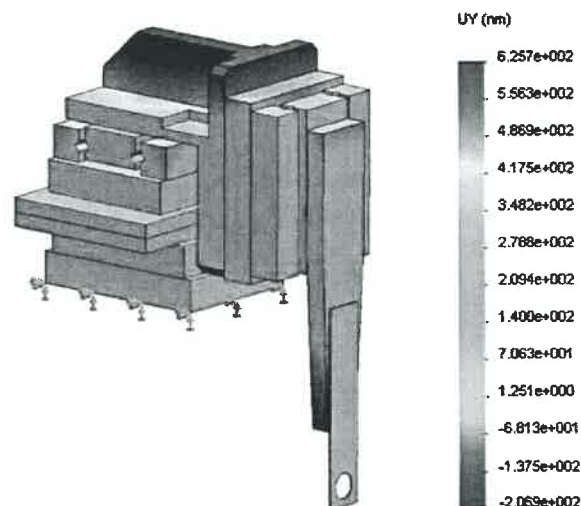


Figure 6. A 3-D FEA model of the Z2-3701 X-Y-Z stages sub-assembly module for thermal drifting in Y direction under a 1 °C temperature variation at y-stage neutral position.

Table 1. FEA results for zone plate thermal drifting in Y direction under 1 °C temperature variation with different Y-stage positions. In this study, the linkage material is SS304 and the adapters material is AL6061.

Y-Stage Position (mm)	Y (nm)
1.0	-6.5
0	-2.3
-0.5	-0.61
-0.6	-0.61
-0.7	-0.17
-0.75	0.061
-0.8	0.24
-1.0	1.4

PRELIMINARY TEST

A test for stacking five zone plates with 80 nm zone width was performed at the APS undulator beamline experimental station 2-ID-D at 25 keV using the Z2-37 stacking alignment apparatus as shown in Figure 7. Figure 8 shows Moire fringes on the scintillator crystal created before final tweaking for the five FZPs alignment. The initial successful x-ray tests demonstrated the performance of the stacking alignment apparatus. Detailed x-ray test results will be published in separated papers [5].

SUMMARY

The mechanical design of multiple FZPs precision stacking alignment apparatus for the

hard x-ray focusing is presented in this paper. FEA results and initial x-ray experimental measurements have shown that the mechanical design of the precision alignment apparatus system will meet design requirements for the hard x-ray focusing in the twenty-nanometer scale. Further x-ray tests for six or more zone plates with 20 nm zone width for higher spatial resolution are in progress.

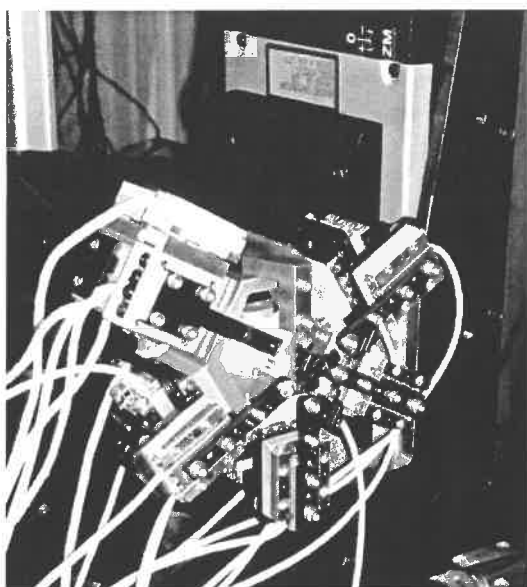


Figure 7. A photograph of a x-ray test setup of the Z2-37 precision alignment apparatus for up to six zone plates intermediate-field stacking at the APS experimental station 2-ID-D. In this setup, the Z2-37 invar base is mounted on a large stepping-motor-driven X-Y-Z stage system associated with a four Z2-3701 X-Y-Z stage module using SmarAct™ SLC-1720S Piezo-motor-driven stages and one set of Z2-3704 X-Y-Z stages module using Micronix™ PPS-20 Piezo-motor-driven stages.

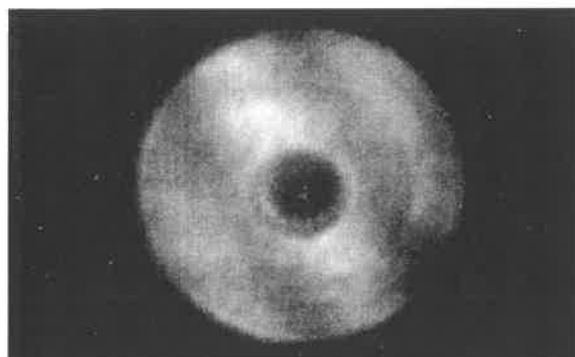


Figure 8. A transmission image on the scintillator crystal with CCD camera showing Moire fringes created before final tweaking for the five FZPs alignment.

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Actuator Design for a High-Speed Large-Range Tip-Tilt-Piston Micro-Mirror Array

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ABSTRACT

This paper presents a new method for the modeling and optimization of actuator performance in a high fill-factor (99%) micro-mirror array design (10,000 1mm^2 mirrors) in order to produce the highest actuation output possible. The high torque output resulting from the optimization enables the mirrors to be driven with three degrees of freedom (DOFs)—tip, tilt, and piston—over large ranges ($\geq \pm 10^\circ$ rotation and $> \pm 30\mu\text{m}$ translation) at high speeds ($\sim 40\text{kHz}$ small stepping rate), all with continuous closed-loop control. The capabilities of this new mirror array will extend the performance of a variety of high-impact technologies including: (i) optical switches, (ii) confocal microscopes, (iii) autostereoscopic display and image capture, (iv) microprojectors, (v) high speed focusable LIDAR, (vi) micro-additive fabrication approaches that utilize principles of light steering (e.g., two photon polymerization or optical tweezers), and (vii) high-powered laser steering systems for defense applications including projectile interception. A seven mirror prototype array has been built to-scale to demonstrate the design's fabrication feasibility.

STRUCTURE

A seven mirror array is shown in Fig. 1 to illustrate the scale and components of the micromirror. Only the middle element is shown fully assembled. The micromirror array is composed of hexagonal unit cells, each of which contains three bipolar electrostatic comb drive actuator paddles, three decoupling flexure linkages and a hexagonal mirror. The actuators are the green/blue paddles which are anchored at two corners to the red actuation plate below and are free to rotate around a single axis as shown in Fig. 2.

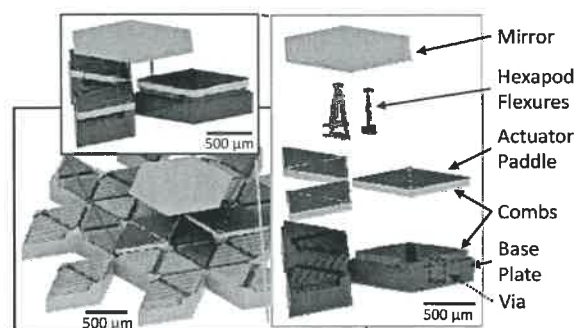


FIGURE 1: Micro-mirror array design.

This single DOF appears as vertical motion to the decoupling flexures anchored near the end of each paddle. These flexures are composed of two intersecting wires that transfer vertical motion, inhibit in-plane motion in one direction and allow rotations of the mirror. The combination of the three decoupling flexure sets acts like a hexapod structure and fully kinematically constrains the mirror. The decoupling flexures transform paddle rotations to mirror rotations via a double transmission which is set by the flexure's location on the paddle.

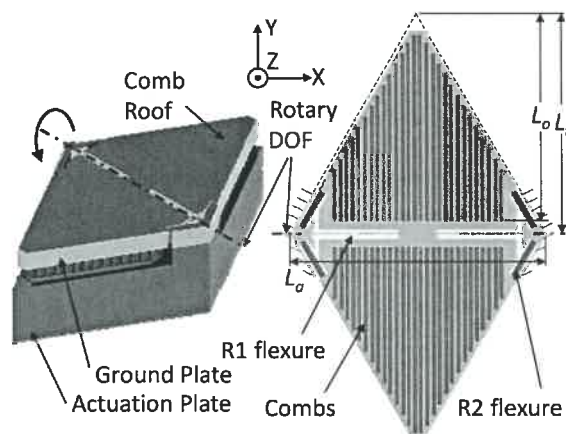


FIGURE 2: Actuator paddle in detail.

¹These are both first authors as their contributions to this paper are equal

Each paddle contains two symmetric sets of combs in order to provide bipolar motion control. The rotary DOF of the actuator paddle is driven by energizing the actuation plate side of one of these comb sets, causing the combs of the (grounded) ground plate to pull down towards the actuation plate. The R1 and R2 flexures of the paddle allow this rotary DOF while resisting all other motion. The design of these flexures (R1, R2) and the anchoring structure around the combs is crucial for resisting comb pull-in (relative X-axis motion) and thus maximizing actuation force.

BACKGROUND

Comb drives have been the focus of extensive optimal design efforts. In particular, the instability due to pull-in has been extensively studied [1–5], and a range of additional effects have been considered including bearing stiffness [2] and varying comb cross-section [3]. These efforts typically focus on deducing a critical stiffness for design [1,2,4,5] and take the comb geometry as given. This work operates in the opposite direction. The optimizer draws upon given flexure bearings which have been designed via FACT to find the comb geometry that maximizes actuation within the given design topology and geometric constraints. The flexure stiffnesses are then adjusted to find the best tradeoff between actuation output and bearing stiffness/strength.

We present a new method to analyze pull-in that extends beyond previous work so as to incorporate both fabrication errors and the stiffness of the full load path loop into closed-form analytical expressions for pull-in displacement. This expression is integrated into a new approach to comb-drive design which uses constraint-based optimization (CBO). CBO offers several advantages when used in design, including rapid redesign, optimal performance, and the ability to study sensitivity to constraint inputs, so as to aid in design decisions.

MODEL ELEMENTS

The multiple elements of the paddles are incorporated into the model via several paths. This includes the combs on the actuation and ground plate- these are able to flex under high voltages with their tips moving in the x-direction towards one another. The remainder of the actuation plate is considered a mechanical ground. The ground plate is analyzed in several

parts. The sidewall linking the comb tips, the roof over the combs and the H-shaped central mass are all considered part of the rigid body of the paddle. This rigid body is able to both translate in X and rotate in θ_z under pull-in loading when one comb set is activated.

The paddle fabrication process involves etching both actuation and ground plate from single crystalline silicon wafers, then aligning the two structures and bonding them together. The alignment error generates an offset, δ , of the combs relative to the ideal equilibrium locations, as shown in Fig. 3.

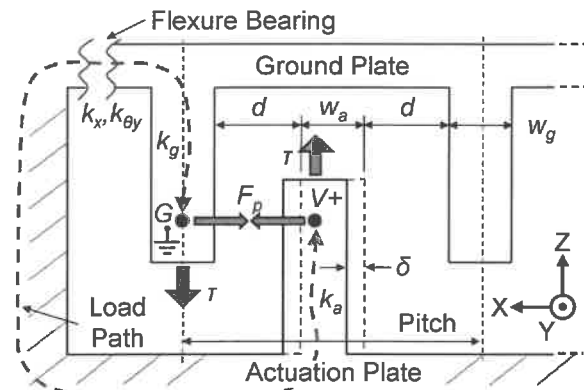


FIGURE 3: Schematic of the comb overlap, showing main geometry, stiffnesses, alignment error δ , and the full load path.

ACTUATION TORQUE

The maximum actuation torque, T_{max} , for the paddles around their rotary DOF is produced by the tangential electromagnetic stress in the overlap area of the ground and actuation plate combs. A relative voltage between these combs, as shown in Fig. 3, drives the ground plate combs downward to overlap further with the actuation plate combs. This force is calculated by determining the actuation stress- the Z-axis force generated per area (pitch*comb length), then integrating the torque per area over the active comb section of half the paddle. The result is shown in Eq. (1), with the variables identified in Fig. 2 and Fig. 3, where L_a is the width of the paddle at the axis- also the side length paddle, $L_t = L_a\sqrt{3}/2$ is the axis-to-tip distance, L_o is the comb overlap length, N is the number of combs on each side, γ_{cf} is the fraction of usable area (≈ 0.825) in the paddle, ϵ_r is the relative permittivity of the air in the gap, ϵ_o is the permittivity of free space, V_{max} is the max voltage applied between the combs which is limited by Paschen's law to be $<300V$ for Si-Si

combs [6] but is typically <200V, d is the nominal gap between comb faces, w_a is the actuation plate comb width and w_g is the ground plate comb width.

$$\tau_{max} = \left(L_t - \frac{2L_o}{3} \right) \frac{L_o N \gamma_{cf}}{2} \frac{\epsilon_r \epsilon_0 V_{max}^2}{d} \quad (1)$$

$$N = \frac{L_a (L_o / L_t)}{w_a + w_g + 2d}$$

The goal of the model is to maximize this expression. The torque scales inversely with d , w_a and w_g , which are the main adjustable parameters for the comb drive. The maximum torque is thus found by thinning the combs. The limit of this is to make the combs so compliant as to cause pull-in during operation. The optimizer must then consider pull-in effects to determine the optimal comb pitch and thickness.

PULL-IN FORCE

The pull-in force, $F_p(x_p)$, is produced by the normal electromagnetic stress in the overlap area of the ground and actuation plate combs. This force incorporates the assembly offset, raised to its maximum value, as shown in Eq. (2), where d_{max} is the maximum average comb overlap length, and x_p is the net pull-in deflection during maximum loading. The expression includes the forces from the combs on either side of the each ground plate comb. When these are precisely balanced and there is no assembly error, the pull-in forces cancel out.

$$F_p(x_p) = \frac{1}{2} \epsilon_r \epsilon_0 L_o d_{max} V_{max}^2 \dots \cdot \left(\frac{1}{(d - \delta - x_p)^2} - \frac{1}{(d + \delta + x_p)^2} \right) \quad (2)$$

LOAD PATH STIFFNESSES

The pull-in force is resisted by the full structural loop stiffness of the actuator, as shown in Fig. 3 which includes the lateral stiffness of the actuation plate combs, k_a , ground plate combs, k_g , the paddle structure stiffness in the X-axis, k_{rx} , and the paddle structure θz rotary stiffness, $k_{r\theta z}$. All of these stiffnesses are functions of the three main comb drive parameters- (d , w_a , w_g). These are the three values to be optimized.

The springs are modeled as series modes whose displacements are summed to determine the net stiffness, k_{tot} , at the midpoint of the ground plate combs, as shown in Eq. (3). The two comb stiffnesses are each exposed to the distributed pull-in load of a single comb, but the two paddle motion stiffnesses are driven by the net total pull-in force. The single comb pull-in force for these motions is scaled by $N\gamma_{cf}/2$ to account for the N other combs, only γ_{cf} fraction of which are usable, each averaging only of $1/2$ the length of the full comb L_o due to the triangular area. The θz term includes an additional lever arm factor, $L_{r\theta z}$ to account for the comb distribution when generating a torque around the Z-axis. This standardizes the input load to all of the springs to be the single comb pull-in force as shown in Eq. (2).

$$k_{tot}^{-1} = k_a^{-1} + k_g^{-1} + \left(\frac{2k_{rx}}{N\gamma_{cf}} \right)^{-1} + \left(\frac{2k_{r\theta z}}{N\gamma_{cf} L_{r\theta z}} \right)^{-1} \quad (3)$$

DISPLACEMENT CALCULATION

The net pull-in displacement x_p is calculated by balancing the load path spring force against the non-linear pull-in force, as shown in Eq. (4).

$$\sum F_x = F_p(x_p) - k_{tot} x_p = 0 \quad (4)$$

The pull-in force generates a 3rd order polynomial, but can be simplified via a Taylor series around the expected operating regime as shown in Eq. (5) with series coefficients t_0 , t_1 , t_2 . This provides a means to bring Eq. (4) back to a closed-form solvable expression which enables the optimization to be run quickly.

$$T \{ F_p(x_p) \} = \frac{1}{2} \epsilon_r \epsilon_0 L_o d_{max} V_{max}^2 \dots \cdot \left(t_0 + t_1 (x - d_1 r_p) + t_2 (x - d_1 r_p)^2 \right) \quad (5)$$

The linearization point for the series is specified as a fraction, r_p , of the initial gap on the smaller side, $d_1 = d - \delta$. In a simple parallel plate actuator (with only two plates), the pull-in threshold would occur at $x_p/d_1 = 1/3$. The addition of the second plate on the opposing side of the moving comb, with distance $d_2 = d + \delta$ adjusts this value. When the assembly error is 0, then the pull-in forces balance and alter the threshold value. As the

assembly error grows relative to the nominal gap, the d_2 term becomes insignificant and the location of the threshold converges back to $1/3$ [2]. The optimization process will tend towards this as smaller gap means more actuation force. A safe design assumption is to assume failure at $r_p \approx 1/3$, so set a maximum value $r_p \approx 1/6$.

The displacement solution is solved using the quadratic formula as shown in Eq. (6). The solution is most accurate around the linearization point $r_p d_1$, which is the target around which the optimization code designs.

$$x_p(d, w_a, w_g) = d_1 r_p - \frac{C t_1 - 1}{2 t_2 C} - \dots$$

$$\frac{\sqrt{(t_1^2 - 4 t_0 t_2) C^2 + (4 d_1 r_p t_2 - 2 t_1) C + 1}}{2 t_2 C} \quad (6)$$

$$C = \alpha_p k_{tot}^{-1}$$

OPTIMIZATION

An optimization is carried out to maximize the actuator performance given the various constraints- including pull-in- on the designs. The pull-in constraint is defined by enforcing the solution of x_p as calculated in Eq. (6) to be at the r_p limit. The three inputs (d , w_a , w_g) are then adjusted to maximize the actuator torque while meeting the gap and fabrication constraints, as shown in Eq. (7).

Given

$$x_p(d, w_a, w_g) = r_p d_1$$

$$d > d_{min} \quad (7)$$

$$(w_a, w_g) > w_{min}$$

Maximize(τ_{max})

The optimization can theoretically proceed over the full three parameter design space, however it can be difficult to converge given the singularity at the $r_p \approx 1/3$ pull-in. The system will always operate along the pull-in constraint boundary as that provides the maximum electromagnetic stress. The pull-in constraint equation can thus be used to define one of the three inputs (d , w_a , w_g) as functions of the other two. A sub-function is used to calculate the nominal gap width d from of the comb widths. This sub-function masks the singularity from the

general torque maximization function. The resulting torque output over the remaining two parameter space is shown in Fig. 4.

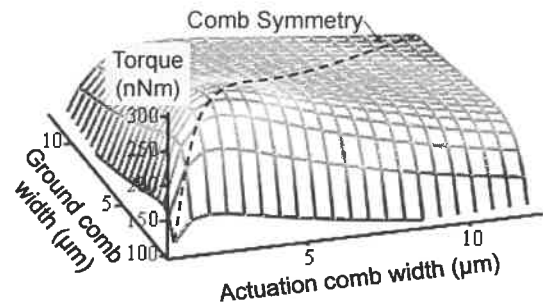


FIGURE 4: Torque output over the two parameter comb width space

The comb widths turn out to have a symmetric input to the performance via the stiffness expressions due to simplifying boundary condition assumptions. This leaves effectively a single variable to scale over in this design for performance maximization. The torque output over a range of equal comb widths is shown in Fig. 5. The solving surface is now smooth with an easily identifiable single peak value occurs at around 3.5 μm .



FIGURE 5: Torque output as a function of the $w_a = w_g$ one parameter space.

CONFIRMATION

The structural kinematics and elastomechanics were confirmed via Finite Element Analysis (FEA) as shown in Fig.'s 6 and 7. The load path stiffness predictions were found to underpredict the overall stiffness by 9%. The slight underprediction is due to the conservative simplifying assumptions of i) no sidewall boundary conditions on the ground plate combs, and ii) comb tip loading. Conservative estimates are desired as they increase the safety factor for the design.

The total deflection of the ground plate paddle under pull-in load is shown in Fig. 6. The maximum deflection occurs about 75% of the way along the comb length. This is due to an interaction of the boundary conditions at the far end of the comb restricting the ground plate

comb deflection, while the θ_z rotation mode appears as a radially increasing deflection with distance from the center of the paddle. The combs were loaded over a trapezoidal area to account for the load distribution under maximum rotation. The far ends of the combs overlap further with the actuation plate combs so see a higher concentration of the pull-in force.

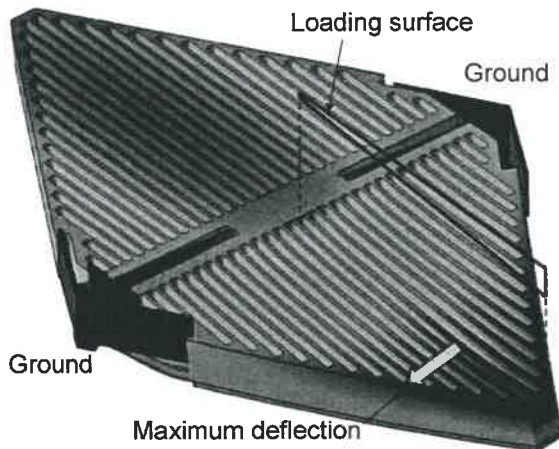


FIGURE 6: Torque output as a function of the $w_a=w_g$ comb widths.

The main DOF stiffnesses of the design were also analyzed in FEA in order to study the effect of the actuation torque on the full design. The R1 and R2 flexures enforce the single rotary DOF of the actuator paddle as seen in Fig. 7. The nearly pure rotation occurring on the paddles drives the three decoupling flexures vertically, controlling the tip, tilt and piston motion of the mirror.

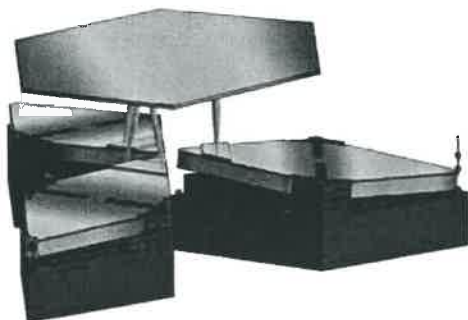


FIGURE 7: FEA confirmation of main DOF.

CONCLUSIONS

A new method is described for the modeling and optimization of electrostatic comb drives in a MEMS micromirror device. This enables the micromirror array to operate at high speeds over large ranges. The new method is able to

incorporate assembly error and the full load path stiffness into the analysis of the actuator performance. The optimization process operates in reverse of the typical analysis, as the overall bearing stiffness is used as an input around which the process optimizes the main comb drive parameters (d , w_a , w_g) to produce the largest possible actuation output. This allows for the actuator to be rapidly resized as the general flexure topology is adjusted, either manually or as part of a larger analysis cycle.

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Flexure Design for a High-Speed Large-Range Tip-Tilt-Piston Micro-Mirror Array

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ABSTRACT

The aim of this paper is to introduce the flexure topology of a new, high fill-factor (99%) micro-mirror array (10,000 1mm² mirrors) that can achieve continuous, closed-loop control of three degrees of freedom (DOFs)—tip, tilt, and piston—over large ranges ($>\pm 10^\circ$ rotation and $>\pm 30\mu\text{m}$ translation) at high speeds ($\sim 40\text{kHz}$ small stepping rate). The flexure topology of this array was designed using the Freedom and Constraint Topologies (FACT) synthesis approach [1], which utilizes geometric shapes to help designers rapidly consider every flexure topology that best achieves a desired set of DOFs. The performance capabilities of this new mirror array will enable, or significantly improve, the performance of a variety of high-impact technologies. Examples include: (i) optical switches, (ii) confocal microscopes, (iii) autostereoscopic displays, (iv) micro-projectors, (v) high-speed focusable LIDAR systems, (vi) micro-additive fabrication approaches that utilize principles of light steering (e.g., two photon polymerization or optical tweezers), and (vii) high-powered laser steering systems for defense applications including projectile interception. A single mirror prototype is currently being built to scale to demonstrate its predicted performance.

CONTRIBUTIONS

In this paper, we introduce a new micro-mirror array (Fig. 1). Specifically, we explain how FACT was used to synthesize the array's flexures such that they (i) only permit the three desired DOFs, (ii) are symmetric and exactly-constrained to achieve maximum precision with minimal parasitic error and thermal drift, (iii) decouple each mirror's actuators to enable independent actuation of the mirrors' desired DOFs over large ranges while accelerating as little mass as possible, (iv) don't permit any under-constrained bodies in the design to protect against unwanted vibrations, and (v) behave as transmission

elements that allow the design to be tuned to achieve a large range of target speeds and strokes for specific applications.

MICRO-MIRROR ARRAY'S FLEXURE DESIGN

The micro-mirror array introduced in this paper is shown in Fig. 1. Although the maximum number of mirrors within the array is limited primarily by the size of the wafer on which the array is fabricated, the array in Fig. 1 shows only seven mirrors. Each mirror in this array is constrained from below by flexures that only permit three DOFs—two rotations (shown as red lines in Fig. 1) and a translation (shown as a black arrow).

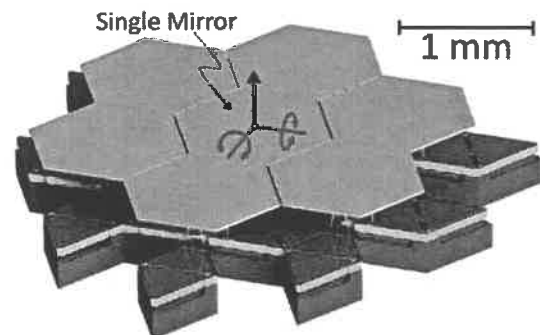


FIGURE 1. New micro-mirror array design

A single hexagonal mirror within the repeating array is shown with different views in Figs. 2A-B. An exploded view of the system is shown in Fig. 2C. The mirror is constrained by three identical axisymmetric serial flexure limbs labeled (1) through (3) in this figure. Each of these limbs consists of two parallel flexure modules stacked in series. One parallel module consists of two wire flexures (colored yellow) that form a triangular truss and join the mirror to one of its three paddles (colored blue and green). These flexures are labeled "Decoupling Flexures" in Fig. 2C. The other parallel module consists of six blade flexures (Fig. 2D) that join the

¹These are both first authors as their contributions to this paper are equal

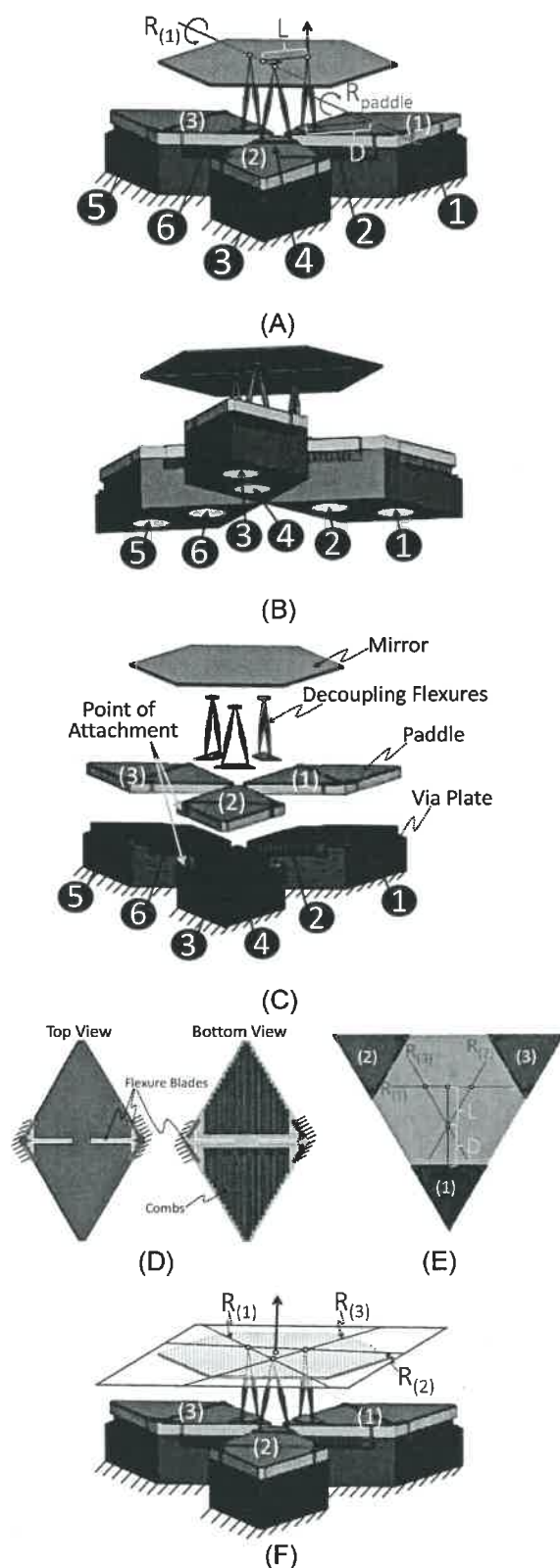


FIGURE 2. Single mirror within the array—top view (A), bottom view (B), exploded view (C), paddle geometry (D), bird's-eye view (E), and freedom space (both active and passive) (F).

corresponding paddle to the fixed via plate (colored red). This second parallel module only permits each paddle to rotate about a single axis located at its center. The paddle on the far right of Fig. 2A, for example, can only rotate about the axis labeled R_{paddle} . Each paddle is driven by a bipolar electro-static rotary comb drive actuator that uses two sets of combs. Each set of combs can be independently charged with a voltage by a corresponding via that passes through the backside of the via plate where controller electronics are attached. The mirror's vias are numbered 1 through 6 in Figs. 2A-C. Note that the electrically charged combs of the via plate mesh and overlap with the combs of the paddles (labeled in the bottom view of Fig. 2D), which are electrically grounded. If the via labeled 1 is given a charge, its corresponding paddle will pull down and rotate about R_{paddle} in the direction shown in Fig. 2A. If the other two paddles are simultaneously held fixed, the mirror will rotate about the axis labeled $R_{(1)}$ in Figs. 2A. To rotate the mirror about the same axis, but in the opposite direction, the via labeled 2 should be given a charge instead of the via labeled 1. Similarly, the mirror can be rotated about the axes labeled $R_{(2)}$ and $R_{(3)}$ (Figs. 2E-F) in different directions by charging vias 3 or 4 that correspond with paddle (2) and vias 5 or 6 that correspond with paddle (3) respectively. Thus, the mirror's six actuation vias can be independently charged to drive any combination of $R_{(1)}$, $R_{(2)}$, and $R_{(3)}$. By charging these six actuation vias different amounts, the mirror can be made to translate along the axis of the black arrow shown in Fig. 2F, and rotate about any axis that lies on the mirror's plane shown outlined red in the figure. This red plane of rotation lines and the orthogonal translation arrow is the system's active freedom space (AFS) [2]. An AFS is a geometric shape that represents all the ways a system's stage (i.e., the mirror in this instance) can be actuated for every combination of actuator loads. Note that the three desired DOFs (i.e., tip, tilt, and piston) shown in Fig. 1 lie within this AFS.

USING FACT TO DESIGN THE FLEXURES

In this section, we utilize the geometric shapes of FACT to explain how the mirror's flexure topology was synthesized. According to FACT, every parallel flexure module (i.e., two bodies directly joined by compliant constraint elements arranged side by side in parallel) possesses a constraint space (CS) and a complementary passive freedom space (PFS). A parallel flexure

module's PFS is a geometric shape that represents all the ways one of the bodies within the parallel module is permitted to move as a result of its compliant constraint elements (e.g., wire or blade flexures) when the other body is held fixed. The module's CS represents the region of space within which every viable compliant constraint element can be placed to permit the motions within its complementary PFS while constraining other unwanted motions. It is important that enough independent compliant constraint elements are selected from a parallel flexure module's CS such that the module will be permitted to move with the motions of its PFS only. If too many compliant constraint elements are selected from the CS such that some are redundant (i.e., they constrain the same motions), the module is over-constrained. The process of appropriately selecting compliant constraint elements from a module's CS such that the resulting module is exactly or over-constrained to achieve the desired DOFs of the module's desired PFS has been addressed in many previous publications [2] and is not the focus here.

Consider the parallel flexure module shown in Fig. 3A that consists of a single paddle joined directly to the via plate by six flexure blades. This module's CS is an infinite number of planes that intersect along a common axis as shown by the blue planes in Fig. 3A. Note from Fig. 2D that two of the paddle's flexure blades lie on the labeled vertical plane in Fig. 3A and the paddle's remaining four flexure blades lie within the labeled horizontal plane. The CS's complementary PFS is a single rotation line (Fig. 3B) that is collinear with the intersection axis of the CS's planes. Thus, the reason the paddle can only rotate about this axis is because a sufficient number of compliant constraint elements were selected from within the module's CS. Now, consider the parallel flexure module shown in Fig. 3C that consists of the mirror joined directly to a paddle by two angled wire flexures. This module's CS is a disk of blue constraint lines that lie on the plane shown. Note that the axes of both wire flexures are coincident with constraint lines within the module's CS. Its complementary PFS (Fig. 3D) consist of (i) a plane of rotation lines that is coplanar with the CS's disk, (ii) a translation arrow that is orthogonal to this plane, (iii) a sphere of rotation lines that intersect at the intersection point of the constraint space's disk, and (iv) screw lines that

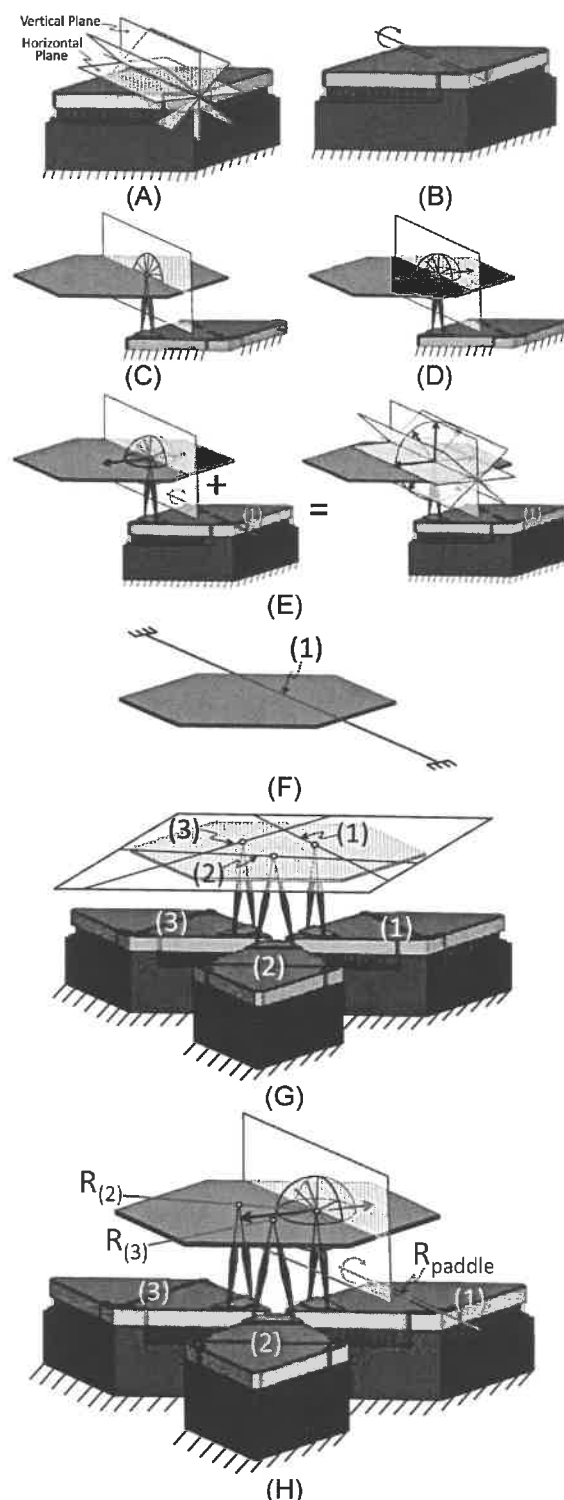


FIGURE 3. Paddle's constraint space (CS) (A) and passive freedom space (PFS) (B), decoupling flexure's CS (C) and PFS (D), limb's effective PFS (E) and effective CS (F), system's effective CS (G), decoupling flexure's PFS satisfies the necessary conditions to correctly decouple the system's actuators (H).

are not shown in the figure to avoid visual clutter. Thus, if the paddle shown in Fig. 3D were held fixed, the mirror would be able to move with all of these motions because enough wire flexures were selected from this PFS's complementary CS.

To determine the effective PFS of the single serial limb shown in Fig. 3E that consists of the two parallel modules shown in Figs. 3B and 3D stacked together, the motions within the PFS of both parallel modules should be linearly combined (i.e., summed together). Thus, if the via plate is held fixed, a mirror constrained by one of these serial limbs could not only move with all the permissible motions within both PFSs, shown on the left side of Fig. 3E, it could also move with every combination of these motions. The resulting PFS, shown on the right side of Fig. 3E, consists of (i) an infinite number of intersecting red planes that contain rotation lines, (ii) a disk of translation arrows that are perpendicular to the axis of intersection of these planes, and (iii) screw lines that are not shown to avoid visual clutter. The complementary CS of this effective PFS is a single constraint line that is collinear with the axis of intersection of the rotational planes of the PFS. Thus, a mirror constrained by a single wire flexure, like the one shown in Fig. 3F, is kinematically equivalent to the serial limb of Fig. 3E (i.e., it does the same job of both constraining and permitting the same motions).

Thus, to determine the effective PFS of the entire single mirror system shown in Fig. 3G that consists of three identical axisymmetric serial flexure limbs arranged in parallel, the effective CS of each limb must first be linearly combined to determine the overall system's effective CS. The complementary PFS of this CS is the effective PFS of the entire system. The effective constraint space of each limb (i.e., a single constraint line) is shown and labeled according to its corresponding limb number in Fig. 3G. The effective CS that results from linearly combining these three constraint lines is the single plane shown outlined blue in the same figure. Its complementary PFS is the red plane of rotation lines and the perpendicular translation arrow shown in Fig. 2F. Thus, if the via plate is held fixed, the overall flexure topology of each mirror within the array of Fig. 1 is correctly designed to passively constrain the mirror to only achieve the motions that result from the combination of the three desired DOFs shown in the same

figure. Note also that this system's PFS is identical to the system's AFS. This means that the actuators have been correctly designed to actively drive all of the system's passive DOFs. Furthermore, note that the fewest number of actuators have been employed to fully actuate the desired system DOFs (i.e., three paddles are used to drive the mirror's three DOFs). Note also that to drive each paddle so that it can rotate in both directions, two sets of combs that are independently charged by two sets of corresponding vias are necessary.

The mirror's flexure topology is also largely symmetric and exactly-constrained to achieve maximum precision with minimal thermal drift and parasitic error over its full range of motion. Since each serial limb within the system contributes a single constraint line (Figs. 3F-G) that restricts a unique DOF, the limbs exactly constrain the mirror (i.e., none of the limbs contribute redundant constraint). Furthermore, the two wire flexures selected from within the constraint space of the parallel module of Fig. 3C are independent and thus exactly constrain the module within each limb. The only source of over-constraint in the design comes from the redundant flexure blades within the parallel module of Fig. 3A. Although, we could have used FACT to exactly constrain this module with five independent constraint lines, such a design would (i) not be able to be made sufficiently stiff to achieve optimal mirror performance, (ii) not lend itself to a practical fabrication approach, and (iii) not be symmetric resulting in harmful thermal instabilities, unbalanced dynamics that induce unwanted vibrations, as well as significant parasitic error as the paddle rotates over its full range. Most of the negative effects caused by the choice to over-constrain this module are mitigated by the fact that many of the redundant blade flexures are made from the same piece of material (i.e., no assembly required during fabrication). Furthermore, these blade flexures will likely avoid appreciable temperature gradients, which are harmful to over-constrained designs, because of their close proximity and high thermal conductivity to thermal expansion ratio (they are made of silicon). Note that if the entire array shown in Fig. 1 is heated, the mirror can only displace along the direction of its translational DOF because of symmetry. This minor potential for thermal drift will not, however, affect the light steering capability of the overall system but

could influence its ability to modulate the light's phase.

The FACT shapes (Fig. 3) used to synthesize the flexure topology of this tip-tilt-piston mirror design also ensure that the design's topology correctly decouples its actuators. Note from Fig. 2A that regardless of how much the paddle in the serial limb labeled (1) is rotated by charging vias 1 or 2 various amounts, the other two paddles remain largely unaffected and experience no harmful forces as the mirror is rotated about the $R_{(1)}$ axis. Thus, the desired mirror DOFs can be independently actuated without requiring the via plate's combs (i.e., the system's actuators) to move during operation, which is the case for stacked actuator designs. Thus, this design achieves high speed in part because its actuators are designed to be stationary and because each paddle can independently actuate the mirror without causing the other paddles to move appreciably. As a result, the amount of mass that is necessary to accelerated to achieve any desired mirror DOF is minimized. This decoupling effect also greatly reduces the system's control complexity. According to FACT, the PFS of a system's decoupling flexures (e.g., the wire flexures shown in Fig. 2C) must satisfy specific conditions to optimally decouple the system's actuators [1]. First, this PFS must not contain the PFS of the actuator's flexure bearing such that the actuator can rigidly pass the intended displacement through the limb to the system's stage. For this design, the PFS of the actuator's flexure bearing is the rotation line shown in Fig. 3B. Note from Fig. 3H that this line, labeled R_{paddle} , is not contained by the PFS of the decoupling flexures (Fig. 3D) so the first condition is satisfied. If this line did belong within the PFS of the decoupling flexures, the mirror would hardly move as the actuated paddle rotates. The second condition states that the PFS of the system's decoupling flexures must contain all the motions caused by the actuation of the other serial flexure limbs so that the system's stage motions induced by one actuator do not affect the other actuators. For this design, note from Fig. 3H that the motions $R_{(2)}$ and $R_{(3)}$, caused by actuating the serial limbs labeled (2) and (3) respectively, both lie within the sphere of rotation lines within the decoupling flexures' PFS of limb (1) thus, satisfying the second condition.

Not only is the flexure topology of this paper's mirror design not over-constrained (except

within the parallel flexure module of Fig. 3A as previously discussed), it is also not under-constrained. A serial limb is under-constrained when any of its stacked parallel flexure modules possess redundant DOFs. Thus, when the stage of an under-constrained system is held fixed, one or more of its intermediate bodies possess one or more permissible motions that are not fully constrained and are thus susceptible to unwanted and uncontrolled vibrations. Note that for this paper's design, if its mirror is held fixed, the three paddles will be fully constrained. Note also that if all the paddles are held fixed, the mirror is exactly-constrained by an effective hexapod. If two paddles are held fixed, the mirror is only free to rotate with the motion caused by the paddle that was not held fixed. Thus, the controller used to dynamically drive this mirror can be simplified and does not need to compensate for poor mechanical design.

The mirror of this is paper's design is shaped like a hexagon for a number of reasons. A circular mirror would be the optimal geometry for a tip-tilt mirror from the standpoint of balanced mass moments of inertia. As long as a circular mirror rotates about an axis that lies on its plane and intersects its central point, the mass moment of inertia associated with that rotation is the same regardless of which axis is actuated (i.e., tip, tilt, or any combination of the two). Unfortunately, however, circles cannot fill space and would thus achieve an array with a low fill factor. Only three repeating regular polygon options exist that fully fill space—equilateral triangles, squares, and regular hexagons. Of these three options, hexagons are the most circle-like in that they possess the most balanced moments of inertias.

Hexagons also enable a significant actuation advantage based solely on their geometry. Regardless of the regular polygon chosen for the mirror within any space-filling array, the maximum area that can be allotted to the bipolar electro-static rotary comb drive actuators is the area of the mirror's polygon itself. Thus, an optimal design should utilize this full area to enable the maximum number of combs that can be packed for achieving the largest possible torque loads to drive the mirror. It is difficult to divide the area under a square into three axisymmetric paddles for driving the mirror's three DOFs. Such a design would require a fourth and thus, redundant actuation paddle to be axisymmetric, which would dramatically

increase the system's control complexity. Although both a triangle and a hexagon can be divided into three such axisymmetric paddles, only the hexagon can be divided into three axisymmetric paddles that utilize the full area of the mirror's polygon and are themselves symmetric about their axis of rotation (Fig. 4). This latter paddle symmetry, like a balanced seesaw, further simplifies the dynamic control of the system. Furthermore, note that a hexagon-mirror's paddles can be made to fully utilize the allotted area of the hexagon in a way that also allows them to trade area real-estate under neighboring mirrors as shown on the right side of Fig. 4. Using this configuration, the paddles can impart a significantly larger torque load to their corresponding mirror because their moment arm lengths are substantially increased. Thus, to optimize the speed with which the mirrors of the design of this paper can be driven, the configuration shown on the right side of Fig. 4 was used.

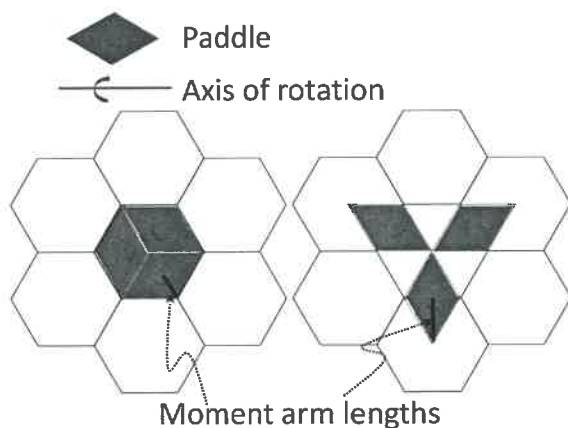


FIGURE 4. Two ways to utilize the full area of a hexagon with three axisymmetric paddles that are also symmetric about their rotational axis. The design on the right side can achieve more torque because of its larger moment arm length.

Finally, note that the flexure topology of the design of this paper possesses a double transmission effect that can be tuned to achieve a large variety of desired mirror speeds and ranges of motion. If each pair of wire flexures, labeled "Decoupling Flexures" in Fig. 2C, are moved toward the center of the mirror such that the parameter labeled D in Figs. 2A and 2E increases and the parameter labeled L decreases the same amount within each of the system's three serial limbs, the mirror's rotational range increases substantially while its

driving torque and speed capability decreases. In like manner, moving these wire flexures away from the mirror's center toward the rotational axis of each paddle (i.e., the mirror's edge for the configuration on the right side of Fig. 4) substantially increases the mirror's driving torque and speed capability while decreasing its rotational range. Thus, by simply arranging the position of the decoupling flexures within the mirror design of this paper, the mirror can be tuned to achieve a large range of speed and stroke combinations that are tailored to satisfy the functional requirements for a multiplicity of diverse applications. Note also that the length along which the decoupling flexures can be displaced is substantially larger for the configuration on the right side of Fig. 4 compared with the configuration on the left side.

CONCLUSIONS

This paper introduces the flexure topology of a new high-fill factor micro-mirror array that achieves unprecedented performance. The array's flexure topology (i) only permits the mirror to passively achieve tip, tilt, and piston DOFs, (ii) is largely symmetric and exactly-constrained, (iii) decouples each mirror's actuators to enable independent actuation of the mirrors' desired DOFs over large ranges while accelerating as little mass as possible, (iv) doesn't permit any under-constrained bodies in the design to protect against unwanted vibrations, and (v) behaves as a double transmission mechanism to enable large ranges of target speeds and strokes for various applications. This work was performed under the auspices of the U.S. Department of Energy by Lawrence Livermore National Laboratory under Contract DE-AC52-07NA27344. LLNL-ABS-658144.

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DESIGN OF A FLEXURE-BASED DYNAMIC-TUNABLE FIVE-AXIS NANOPOSITIONER

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In this paper, we present the development of a flexure-based five-axis nanopositioner with dynamic-tuning capability for parallel nanomanufacturing applications, e.g. electro-machining (nano-EM), dip-pen nanolithography and nano-scratching. Static, dynamic, and dynamic-tuning experiments were devised and performed to characterize the range, speed, and resolution characteristics of the nanopositioner.

DYNAMIC-TUNING FOR NANOPOSITIONER

Nanopositioners can move objects of different sizes with nanometer precision. They are important as they set the limits on our ability to measure, manipulate, and manufacture physical systems. To fulfill the stringent repeatability and precision requirements, compliant mechanisms, i.e. flexures, are often used for their advantages (no wear between joint members and free of backlash etc.) over traditional mechanical linkages [1].

It is known that the dynamic performance of a flexural mechanism depends on its natural frequency; however, this is often compromised by the required stroke of the mechanism. In other words, high natural frequency can only be achieved at the expense of reduced stroke. In this work, we aim to develop a dynamic-tunable flexure-based nanopositioner that allows trade-offs between its speed (natural frequency) and range (stroke)—a concept inspired by compliant actuators used in humanoid robots [2].

The dynamic-tuning effect is achieved by exploiting the "stress-stiffening effect", i.e. the stiffness of a beam increases when it experiences tensile loads in the axial direction [3]. The natural frequency of a simply supported beam with axial force N can be described by Equation (1), where ω is the natural frequency; l , A , I , E , ρ is the length, cross section area, moment of inertia, Young's modulus, density of the beam, respectively [3]. For example, the natural frequency of a 70mm long and 1mm

thick titanium beam can be shifted from 450Hz to 1164Hz when 1000N tensile force is applied. Note that compression load is not used as it may cause buckling and instability.

$$\omega = \left(\frac{\pi}{l}\right)^2 \sqrt{\frac{EI}{\rho A}} \sqrt{1 + \frac{N^2}{\pi^2 EI}} \quad (1)$$

Figure 1 presents the simulated results of a symmetric double parallelogram (DP) mechanism with increasing axial loads until the yield stress is reached. Figure 1A shows the CAD model. The material used in the FEA simulation is titanium.

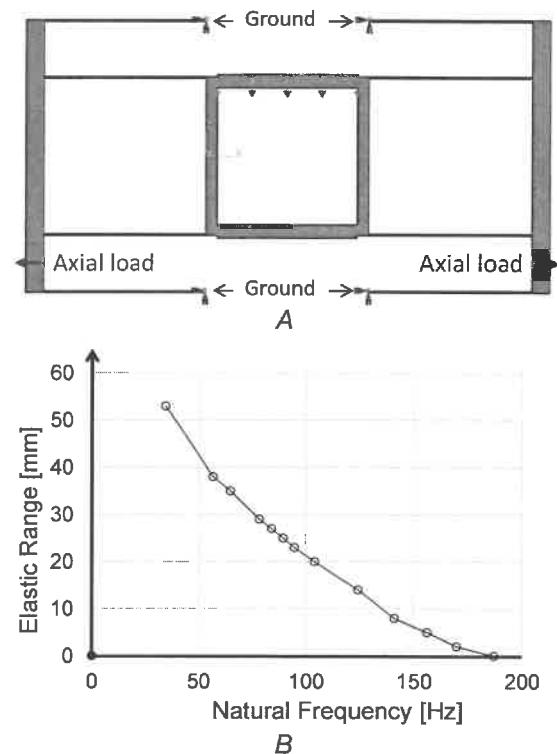


FIGURE 1. Simulated results of a symmetric DP flexure using Abaqus 6.12. A: CAD model with load and boundary conditions; B: The relationship between range and frequency

NANOPOSITIONER DESIGN

Figure 2 shows the layout of the nanopositioner, where an in-plane X-Y stage and an out-of-plane $Z-\theta_x-\theta_y$ stage are connected in series to achieve five degrees of freedom. The design of the X-Y stage utilizes the principle of constraint-based design [4] by using multiple folded-beam mechanisms to decouple the X and Y motion. The extended range in the X and Y directions are achieved by the symmetric DP mechanism. To implement dynamic-tuning, two piezoelectric actuators (green arrows in Figure 2; PI P-845.20) are used in each DP mechanism to generate axial loads (stress) for dynamic-tuning. The dynamic-tuning design is implemented on the X-Y stage to enable trade-offs between its range and speed.

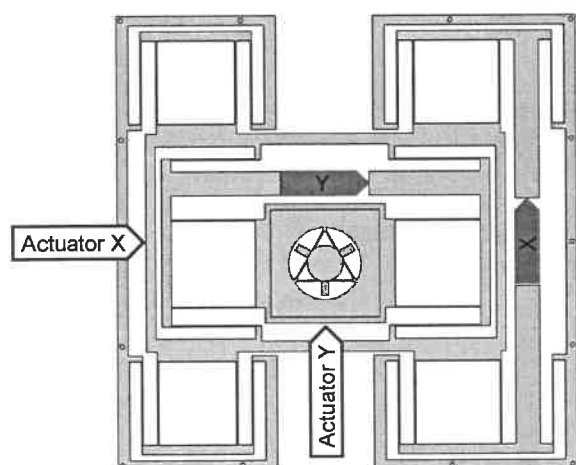


FIGURE 2. Layout of the nanopositioner. Green arrows indicate the piezoelectric actuators (PI P-845.20) for stiffness-tuning; white arrows indicate the linear actuators (PI M-230.10) for in-plane actuation.

Figure 3 shows the layout of the out-of-plane compliant stage which generates independent motions in the Z, θ_x , and θ_y directions. The central stage is supported by six flexural arms. Three piezoelectric actuators (Newport NPA100) are installed below the motion tabs extending from the central stage, as indicated by the blue circles. Three capacitance probes (in red; Lion Precision C30/CPL290) are used to measure the out-of-plane displacements for closed-loop position control. Piezoelectric actuators are used for the out-of-plane stage for high-speed parallelism control and nano-patterning application.

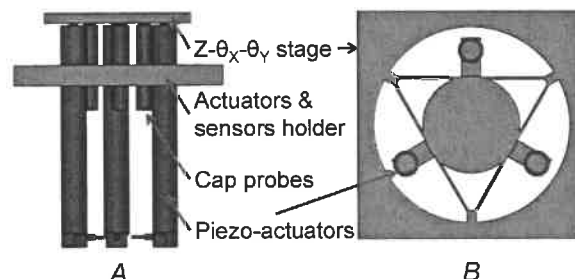


FIGURE 3. $Z-\theta_x-\theta_y$ stage with integrated actuators and sensors (Newport NPA100 and Lion Precision C30). A: Side view; B: Top view

ASSEMBLY AND CHARACTERIZATION

Figure 4 shows the five-axis nanopositioner prototype with integrated actuators and sensors.

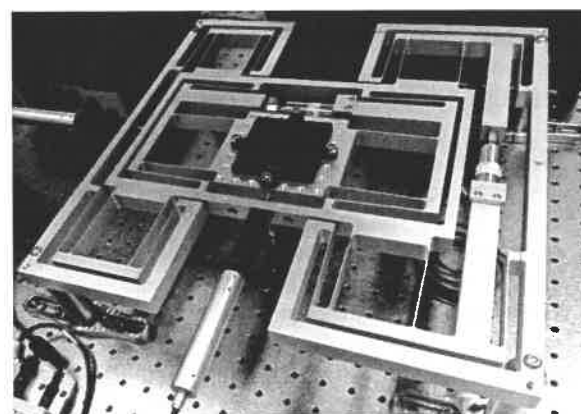


FIGURE 4. Prototype of the five-axis nanopositioner

To characterize the dynamic properties of the nanopositioner, we first conducted an impulse response experiment to obtain resonant frequencies of the nanopositioner in different axes. Resonant frequencies were calculated from the displacement data by performing the fast Fourier transform (FFT). Table 1 summarizes the measured and simulated resonant frequencies. These parameters are used in the dynamic-tuning experiments as well as the implementation of closed-loop control.

TABLE 1. Natural frequencies of the five-axis nanopositioner

Freq [Hz]	X	Y	Z	θ_x	θ_y
Simulated	15.5	20.1	2236	3499	3558
Measured	14.5	20.3	1936	3223	3182

To understand the level of cross-axis motion coupling and implement the open-loop control, static displacement experiments were performed

to identify the input-output matrix S_X that maps actuator displacements to mechanism displacements. Equations (2) and (3) [5] are used to obtain the matrix S_X . The measured results are shown in Equation (4).

$$\begin{aligned} X_C &= S_X X_A \rightarrow [\Delta x \ \Delta y \ \Delta z \ \Delta \theta_x \ \Delta \theta_y]^T \\ &= S_X [x \ y \ z_1 \ z_2 \ z_3]^T \end{aligned} \quad (2)$$

$$X_A = S_X^{-1} X_C \quad (3)$$

$$S_X = \begin{bmatrix} 0.950 & -0.011 & -0.002 & -0.001 & -0.002 \\ -0.008 & 0.987 & -0.038 & -0.033 & -0.002 \\ 0.002 & 0.001 & 0.105 & 0.157 & 0.108 \\ 0 & 0 & 0 & -0.013 & 0.010 \\ 0 & 0 & -0.011 & 0.004 & 0.007 \end{bmatrix} \quad (4)$$

POSITIONING EXPERIMENT

First, static positioning experiments in selected axes (X, Z, and θ_x) were performed to demonstrate the range and precision of the nanopositioner. In these experiments, the input-output matrix, S_X , was used as an open-loop controller. Figure 5 shows the measured displacements and rotation in the X, Z and θ_x axis respectively. The displacements and rotation are plotted versus open-loop displacement commands in the left column, and the off-axis errors are plotted versus displacement commands in the right column. Overall, we found the off-axis parasitic motions to be reasonably small, i.e. 0.05% of the device's range.

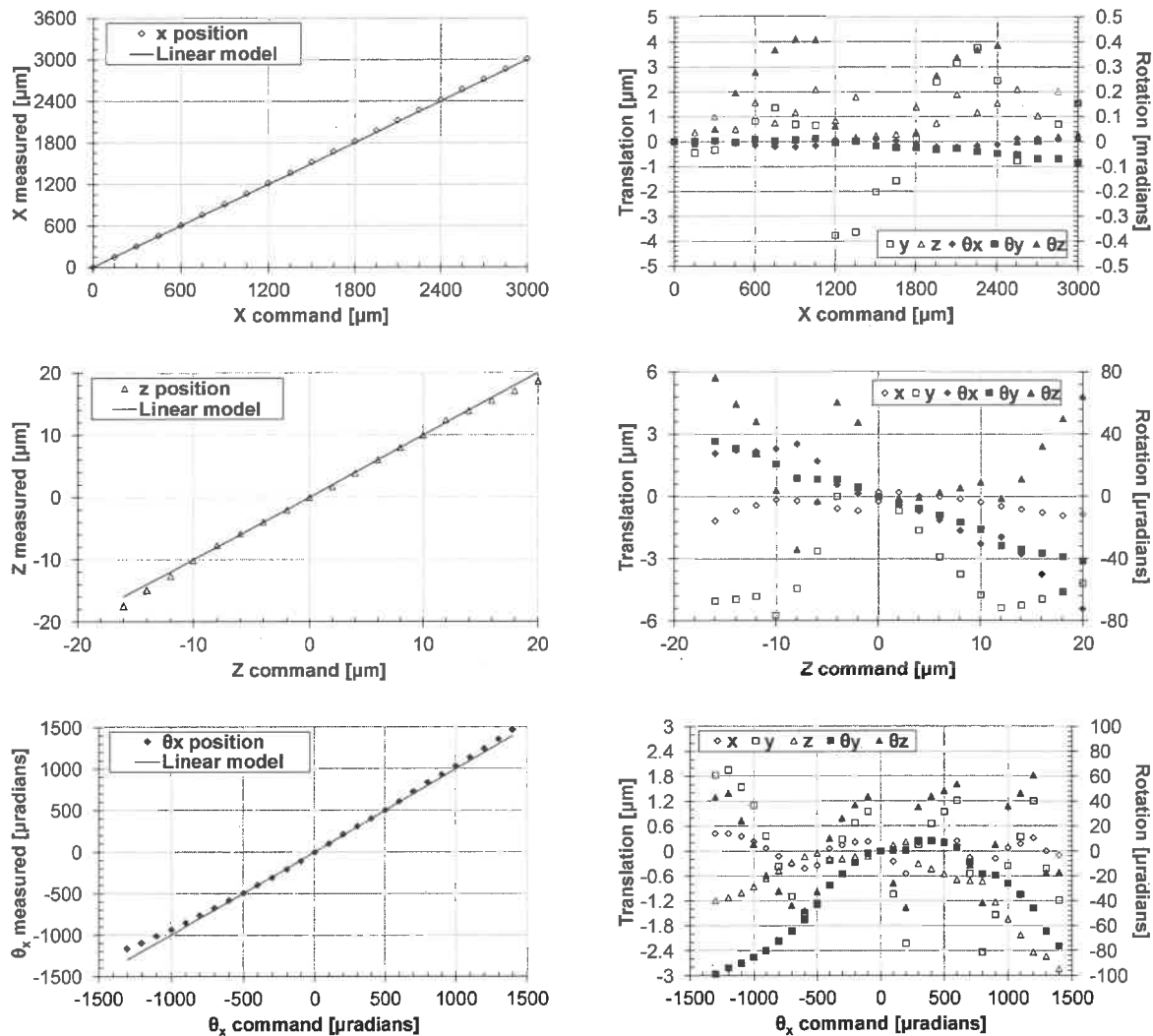
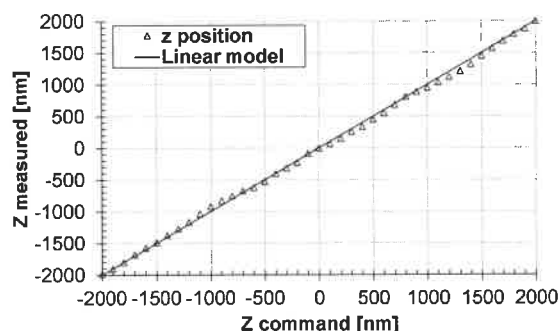
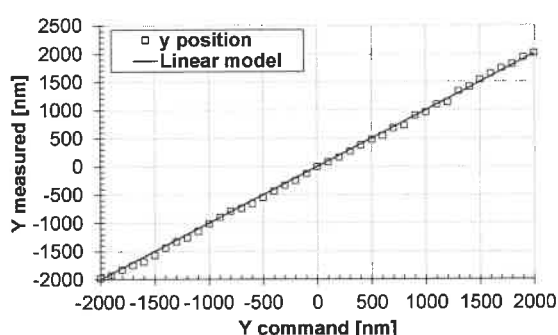


FIGURE 5. X, Z, and θ_x position results

In the second experiment, the nanopositioner was commanded to generate precise X and Z motions with the open-loop controller in a local work volume to demonstrate its nanoscale positioning capability. Figure 6 plots the measured displacements (left) and off-axis errors (right) versus open-loop displacement commands. The results demonstrate the nanopositioner can precisely perform multi-axis nano-patterning tasks with 10s nanometer precision. The precision can be further improved with closed-loop control.



DYNAMIC-TUNING EXPERIMENT

To validate the in-plane dynamic-tuning capability, axial loads (displacements) were applied to the X-Y stage by the stiffness-tuning actuators shown in Figure 2. The results are shown in Figure 7, where resonant frequencies in X and Y directions can be increased by 2-3 times with increasing axial loading. The measured resonant frequencies are constantly lower than the simulated results and the errors increase with increasing displacements. This may be due to the following reasons: (1) misalignment between the piezoelectric actuator and the loading axis and (2) imperfect actuator constraints.

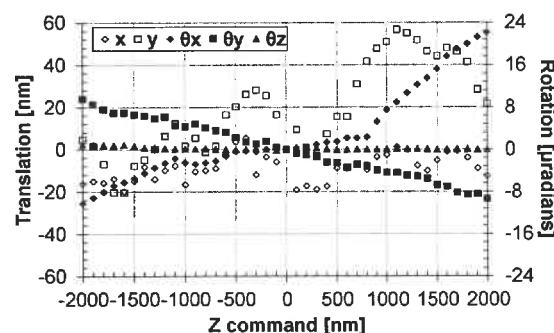
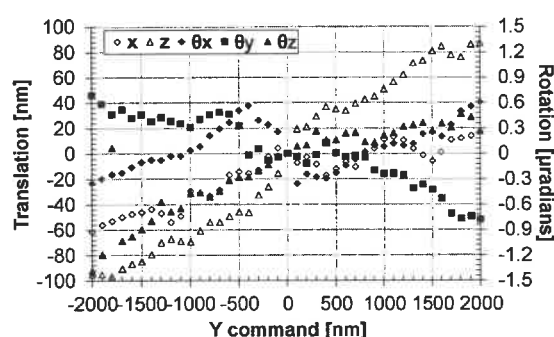


FIGURE 6. Y and Z position results

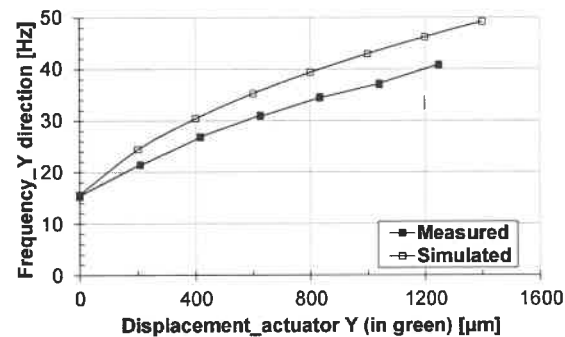
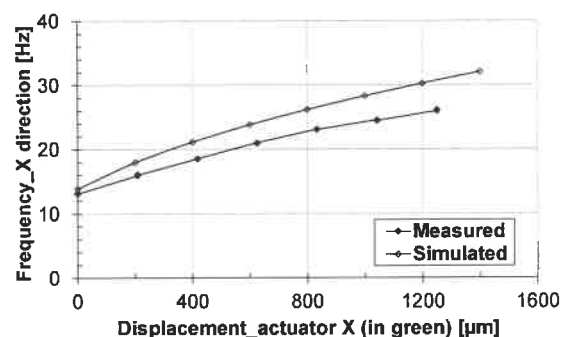


FIGURE 7. The relationship between resonant frequency and applied displacement

NANOMANUFACTURING STATION

Figure 8 presents the conceptual design of a nanomanufacturing station design that includes the five-axis nanopositioner integrated with a custom-built AFM tip/stage assembly, where an AFM tip array is affixed to a manual Z-positioner with a side imaging camera that provides coarse distance control between the AFM tips and the sample. During a nano-patterning process, the $Z-\theta_x-\theta_y$ stage controls both the Z position of the tip array and the parallelism between the array and the sample, while the X-Y stage performs the actual X-Y scanning and patterning.

When the area to be patterned is smaller than the default range of the X-Y stage, the stiffness-tuning commences. According to the FEA results in Figure 1B, the natural frequency of the X-Y stage can be easily increased by a factor of 2 to achieve higher patterning speed and better dynamic characteristics.

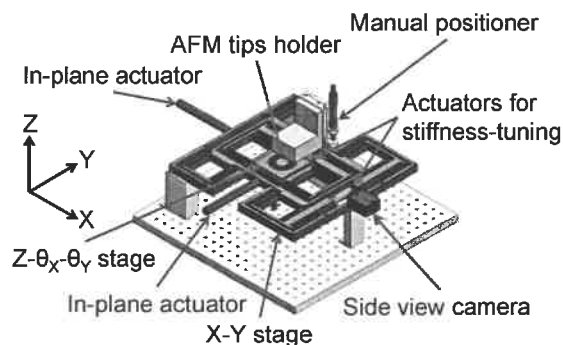


FIGURE 8. CAD model of the nanomanufacturing station

CONCLUSION

We have developed a dynamic-tunable five-axis nanopositioner. In this paper, we presented the design, modeling, assembly, and characterization of the nanopositioner, including the static, dynamic, and frequency-tuning characteristics. Our experiments indicate the nanopositioner can control the stage position within $\pm 25\text{nm}$ and tune its natural frequency between 15 to 40 Hz, which proves effective trade-offs between the natural frequency and elastic range. The demonstrated range and precision make the five-axis nanopositioner suitable for various large area parallel nanomanufacturing applications, including nano-EM, dip-pen nanolithography and nano-scratching. A parallel nanomanufacturing station will be developed based on the nanopositioner and a custom-developed AFM.

ACKNOWLEDGEMENT

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DESIGN OF A MICROFLUIDIC DEVICE TO EXTRACT DYES FROM FIBERS FOR FORENSIC ANALYSIS

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INTRODUCTION

Due to the quantity of textile materials in the environment, there is a high probability of fiber transfer during the commission of a crime. As a result, identification of fiber samples often plays a critical role in criminal investigations. In a typical case, a sample of known origin must be compared to an evidence sample to determine if both could have the same origin. A wide variety of techniques are available to test this hypothesis and each provides a different type of information. Forensic fiber examinations begin with non-destructive visual comparison of characteristics such as color, diameter, cross-sectional shape, birefringence, refractive index and fluorescence. These techniques include light microscope, UV-visible spectrometry and Fourier Transform Infrared spectrometry.

One way to add additional specificity is to identify the dyes used to color the fabric. Such analysis are more sensitive but destroy the evidence sample. Among the quantitative techniques available are high performance liquid chromatography, capillary electrophoresis, pyrolysis-gas chromatography/mass spectrometry and time-of-flight mass spectrometry. The latter technique is becoming increasingly popular for identifying the dye material from a fabric with a high degree of molecular specificity. It can measure the mass of a distribution of different dye molecules extracted from the fiber sample.

However, to use these analytic techniques, dye molecules must be separated from the fiber sample. While useful and pertinent research has been conducted to optimize macroscopic extraction conditions, what is needed is a microfluidic device that can quickly and automatically extract the dye from the fabric. The resulting dye solution could then be loaded into an LC column for separation and subsequent detection by mass spectrometry. Such a system would make it feasible for the forensic examiner to apply the quantitative methods to a larger number of samples while minimizing the risk of human error and sample contamination. Microfluidic systems have been developed for a variety of applications from chemical analysis to

DNA amplification. The differences in the dye extraction problem are 1) the requirement to insert a macroscopic solid object into the device and 2) aggressive extraction solvents such as pyridine that limit the compatible materials.

This paper describes the development of a system to automate the dye extraction process. The system uses a microfluidic cavity to separate the dye from tiny fiber strands less than 3 mm in length and 50 μm in diameter and then prepare the dye for MS analysis – all in less than 10 minutes. This automation is less labor-intensive and allows the forensic examiner to analyze multiple samples at once. Additionally, the system could provide a higher sample concentration than can be achieved manually, thus reducing the size of fiber needed to make a positive ID.

EXTRACTION SYSTEM

The process begins with the forensic examiner placing a small fiber sample onto a flexible microfluidic cavity (10 μl volume) shown in Figure 1. This is typically done using a microscope and tweezers. The examiner then seals the chamber by placing the glass cover over the cavity and checks that the fiber is clamped in place. This design feature allows the examiner to see the fiber both before and after the process.

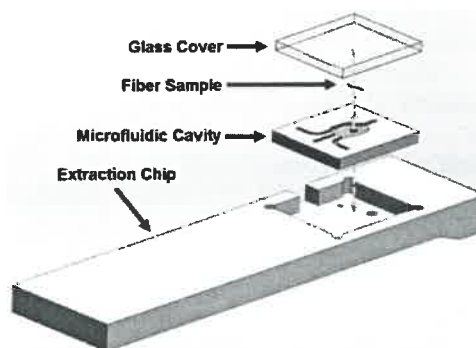


FIGURE 1. The fiber sample is placed on the molded FFKM microfluidic cavity, sealed with a glass cover and inserted into the extraction system.

A significant design challenge was finding a material that would seal the cavity as well as stand up to the pyridine solvent. The solution was to use

FFKM – often called elastomeric Teflon – to create the extraction cavity and sealing it with a glass cover. The FFKM cavity was machined with a 1/32" milling tool operating at 30,000 rpm for these prototypes but will be molded in future versions.

Dye Extraction

The extraction chip holding the sealed microfluidic cavity is then inserted in to the system and fits under the heater at the bottom of Figure 2.

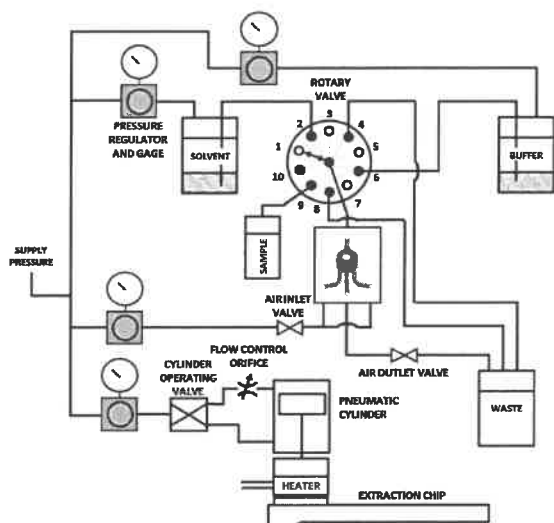


FIGURE 2. Extraction system layout.

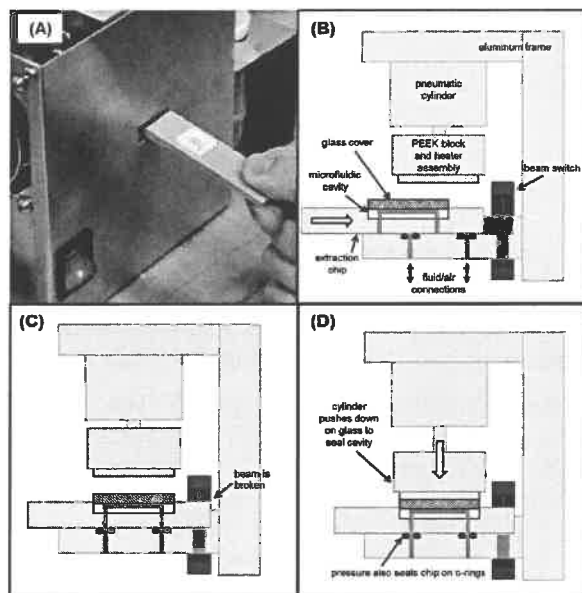


FIGURE 3. The chip is inserted into the system (A-C), breaking the beam of an optical switch. (D) A pneumatic cylinder is triggered, providing sealing pressure to the cavity and fluid connections.

Figures 2 and 3 show the components and steps in the extraction process. When the chip is inserted into the system (3A), it breaks the beam of an optical switch and triggers a pneumatic cylinder to seal the cavity (Figure 3A-3C). The cylinder loads the glass against the flexible cavity to seal the cavity

as well as the fluid connections through the bottom of the extraction chip (Figure 3D).

Fluid, air, waste, and extracted dye are routed to and from the microfluidic cavity through shut-off valves and a 10-position rotary valve shown in Figure 2. Pressurized reservoirs supply solvent (a mixture of pyridine and water) into the cavity to immerse the fiber.

A resistive heater in the clamping block heats the solvent through the glass cover to produce solvent temperature of 90°C in less than 50 sec. High temperature reduces the extraction time of the dye.

Evaporation

After the dye is extracted, the pyridine is evaporated by flowing air through the cavity. This process was also a challenging aspect of the design. The number and shape of the air passages and the air volume/speed were instrumental in creating controlled evaporation of the solvent without blowing the mixture out of the chamber. For a fluid temperature of 90°C, the evaporation process is completed in 1 minute.

Re-dissolution

After the pyridine is evaporated, the dye molecules coat the cavity walls. A buffer solution is introduced into the cavity to re-dissolve the molecules. The temperature of this process needs to be controlled for best results. The buffer solution and reconstituted dye molecules are then pumped through the inlet to a sample container or directly into an HPLC-Mass Spectrometer for analysis. Waste fluid and solvent vapors are pumped to a waste container that is vented by an activated charcoal filter.

RESULTS

Figure 4 shows a red-dyed fiber in the microfluidic cavity before and after extraction with the device.

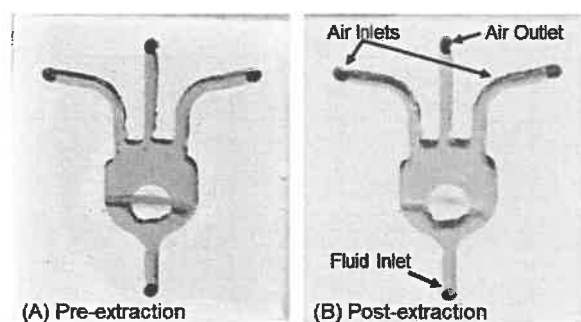


FIGURE 4. Microfluidic cavity and red fiber (A) before and (B) after dye extraction.

The device has been used to successfully extract direct dyes from cotton threads (a bundle of fibers) and acid dyes from nylon threads as small as 3 mm in length. Timing of each extraction step was as follows:

1. Solvent introduction and heating for extraction → 3-5 min.
2. Evaporation of solvent → 2 min.
3. Waste ejection and buffer introduction → <1 min.
4. Re-dissolution of dye and ejection of sample → 1-2 min. (longer for larger sample volume)

HPLC and MS Measurements

High Pressure Liquid Chromatography (HPLC) and Mass Spectrometry (MS) were used to analyze the

red dye extracted from the fiber sample. The HPLC separates the molecules based on polarity and MS measures the mass/charge ratio. The results for the extracted red dye are shown in Figure 5. The retention and EIC time from the HPLC Fig 5(A) identify a region of interest and the charge ratio (m/z) of 631.08 from Fig 5(B) identify the DR 81 red dye molecule. This extraction system was successfully employed with HPLC-MS analysis for the detection of the dyes listed in Table 1.

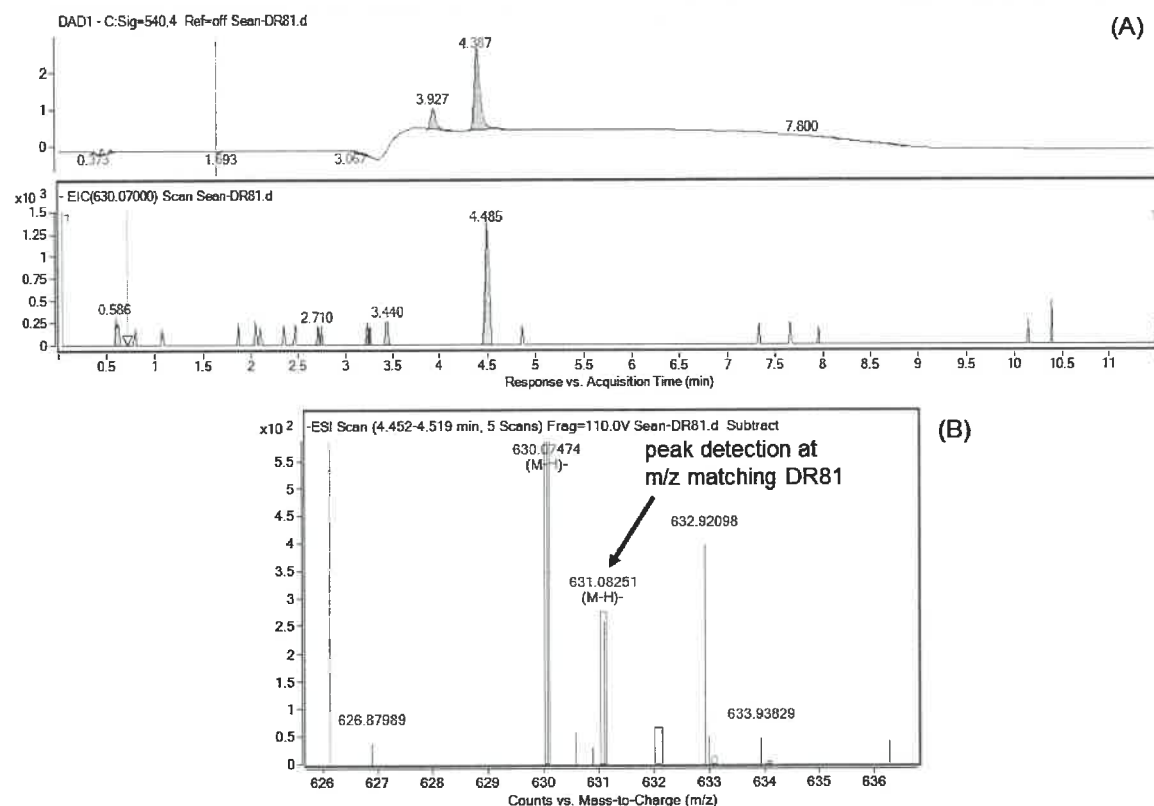


FIGURE 5. Results of analysis of Direct Red 81 ($C_{29}H_{21}N_5O_8S_2$) extracted dye sample. (A) The retention time and EIC time from LC analysis and (B) the mass spectrum show the detection of the DR81 dye.

TABLE 1. List of extracted dyes

Color Index Name	Fiber Material	m/z
Direct Green 26	Cotton	544.08
Direct Red 81	Cotton	314.53
Direct Yellow 106	Nylon 6,6	298.99
Acid Blue 25	Nylon 6,6	393.05
Acid Yellow 49	Nylon 6,6	424.00
Acid Red 186	Nylon 6,6	514.33
Acid Red 114	Nylon 6,6	785.10
Acid Yellow 151-9	Nylon 6,6	807.07

Direct Injection to MS

The direct coupling of the extraction system output to a Quadrupole Time of Flight - MS (QTOF-MS) device significantly improves the

limits of detection for the system. The QTOF-MS and the dye extractor are shown in Figure 6. Acid Blue 25 dye was extracted from an individual nylon fiber (< 1 ng) less than 10 mm in length. The sample was sent directly to a mass spectrometer from microfluidic system, where the characteristic m/z (393.05) of the ions was detected.

In addition to extraction from individual fibers, the system can be used to extract from multiple fibers with different dye colors simultaneously. Acid Blue 25 and Acid Yellow 49 dyes were extracted from individual nylon fibers in a single test where direct injection to the mass spectrometer was used. The mass spectra of the individual dyes is easily distinguished in Figure 7.

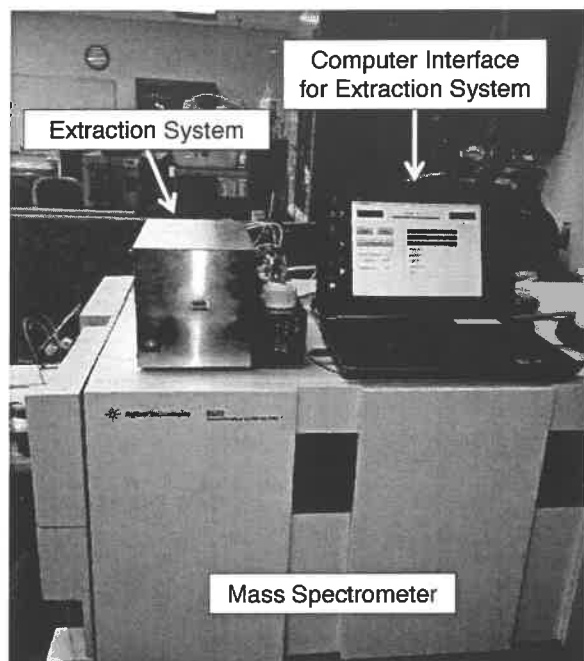


FIGURE 6. Direct coupling of extraction system to mass spectrometer.

CONCLUSIONS

Automated dye extraction from trace fibers in a microfluidic system has been achieved. The extraction process is accomplished in under 10 minutes with minimal fabric and chemical volume. While the extraction is taking place, the forensic examiner is free to prepare the next sample for extraction in a parallel machine, thus increasing throughput and addressing the back-log of samples currently awaiting analysis within the justice system.

Future work will push detection limits to their maximum by minimizing fluid volume to extract dye from even smaller samples. Furthermore, solid-extractions applications are being developed for drug testing, hair analysis, food quality control, etc. Since the device provides control over operation conditions (temperature and process times), extraction can be optimized for a variety of dye and fiber types, as well as other solid-extraction applications.

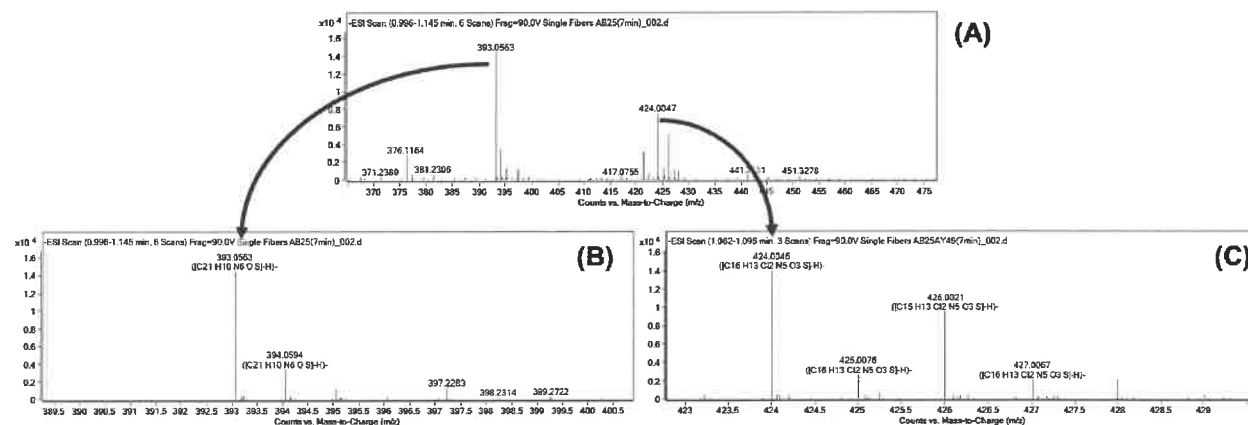


Figure 7. (A) Mass spectra of Acid Blue 25 and Acid Yellow 49 dyes from combined extraction (individual fibers 5-10 mm in length). The individual mass spectra of (B) Acid Blue 25 and (C) Acid Yellow 49 are easily distinguishable.

INTERFEROMETRY AND OPTICS AS EMPLOYED IN ADVANCED LIGO

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SUMMARY

A second generation of gravitational-wave detectors employing laser interferometric sensing of test-mass motion is currently in commissioning. The basic notion is to use a Michelson interferometer to sense the strains in space-time due to a passing gravitational wave. Due to the very small anticipated signal sizes, the actual instrument must have superb optics, have km-scale arms for sensitivity, and perform as limited by quantum and thermodynamic fundamental limits. The design drivers, realization, and the performance to date of both the components and the overall system will be described.

WIDE-FIELD INTERFERENCE MICROSCOPY FOR AREAL TOPOGRAPHY OF PRECISION ENGINEERED SURFACE

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INTERFEROMETERS FOR SURFACE ANALYSIS

Interference microscopy is a well-established optical method for areal surface topography and characterization. Of the available techniques, the dominant method for general applications in precision engineering is coherence scanning interferometry or CSI, because of its ability to measure a wide variety of surface textures. CSI microscopy relies on interference fringe contrast, optionally combined with interference phase, to determine surface heights [1-3].

MEASUREMENTS OVER A WIDE FIELD OF VIEW

Although the range of part sizes encompassed by "microscopy" is subject to interpretation, most often the field of view of microscopes is limited to low magnifications (e.g. 2X) and a 10 mm field of view. Many precision-engineered parts are much larger than this while still requiring the precision of interferometric metrology.

For fast and convenient measurements of surface form and waviness, there is an interest in extending the basic field of view beyond the usual limits of microscope objectives. Some solutions rely on a change in technology to grazing incidence [4], geometrically desensitized interferometry [5], or infrared interferometry [6]. These solutions trade some of the attractive performance benefits of CSI for a larger field of view, and also restrict functionality to form only, whereas detailed surface texture is accessible with CSI using higher magnifications.

There is therefore an interest in extending CSI to larger fields of view. In a previous ASPE paper, we discussed several of the obstacles to this extension, including the optical configuration, the ease of use and the acquisition and processing of interference signals that may be weak and difficult to interpret at the low numerical aperture (NA) values typical of low magnification objectives [7]. Here we report significant new progress in hardware and software for wide field of view areal surface topography metrology for precision engineering.

NEW INTERFEROMETER DESIGN

Researchers and manufacturers have developed a number of configurations based on dedicated systems for wide field of view CSI systems. Most of these systems are of the Twyman-Green or Michelson geometry [6, 8] and are dedicated to low magnification measurements of form.

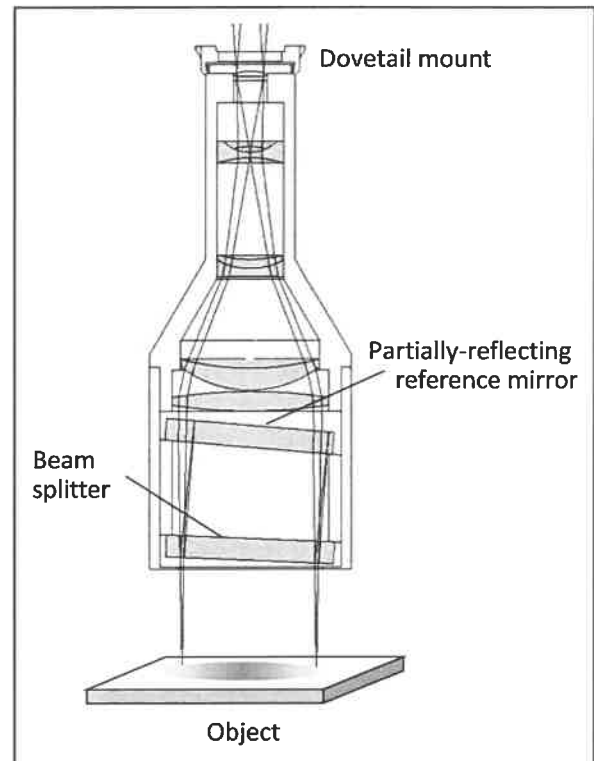


FIGURE 1. Wide-field of view 0.5x objective [9].

We propose an approach based on new, compact and removable low-magnification 0.5X, 1X and 1.4X objectives. This approach allows for form and fine-scale texture measurements up to 100X with a single, multi-purpose platform. FIGURE 1 illustrates the new design for wide-field objectives using a plate beamsplitter and partially-reflecting plate reference mirror [9, 10]. This interferometer realizes an equal-path geometry without the large cube beamsplitter

and reference arm of the more familiar Michelson configuration. The partially transparent surfaces obviate the need for the central obscuration of the Mirau type objective, while retaining the advantages afforded by placing all of the optical components along the same axis. The beam splitter and reference surfaces are oriented so as to reject unwanted reflections, resulting in high fringe contrast with minimal scattered light. Although the imaging is through tilted plates, the optical design achieves well-controlled lateral chromatic aberration and distortion $< 0.1\%$ over the full field at 0.015 NA. The complete objective is significantly lighter and more compact than traditional wide field objectives.

A dovetail mount allows for installing the new objective interchangeably with objectives of other magnifications on a complete CSI system for both form and texture measurements (FIGURE 2). The 0.5X wide-field objective is matched to appropriate tube lenses and camera formats to maximize the field of view. A megapixel camera provides lateral sampling of $32\mu\text{m}$ over a $32 \times 32\text{mm}$ area.



FIGURE 2. Multi-functional interference microscope system with interchangeable objectives, here configured with the 0.5X objective.

DATA ACQUISITION AND PROCESSING

At low magnifications, the range of random surface heights present within the point spread function of the optical system can be comparable to the wavelength of light when measuring rough parts, resulting in lost fringe contrast [11]. This effect combines with the large variations in surface texture and effective reflectivity of common precision-engineered

parts to create significant challenges in signal detection.

Advances in data collection and processing since our last report have dramatically improved the data density on difficult surfaces [12-14]. The result has been an improvement in the number of useable height values when compared to conventional CSI in some cases from 30% to 99%. In many cases, the system reports full 3D topography maps for parts that are not discernable in the intensity images because the surface texture scatters most of the incident light.

The platform of FIGURE 2 offers user-selectable measurement modes, depending on the desired speed, surface height range and measurement precision. These modes include CSI with frequency-domain data processing [15], sub-Nyquist sampling for 3X higher data acquisition speed, and a mode that combines CSI with sinusoidal phase shifting interferometry (PSI) for low noise on smooth parts[14, 16].

Measurement precision depends on part texture, with best results for optically smooth surfaces. The surface topography repeatability [16] for a single image field on a polished reference artefact is 2 nm for individual CSI measurements and $0.1\text{nm}/\sqrt{\text{Hz}}$ for continuous averaging using sinusoidal PSI [14]. This low noise level enables measurements of fine detail even on reference specimens such as the SiC flat, as shown in FIGURE 3.

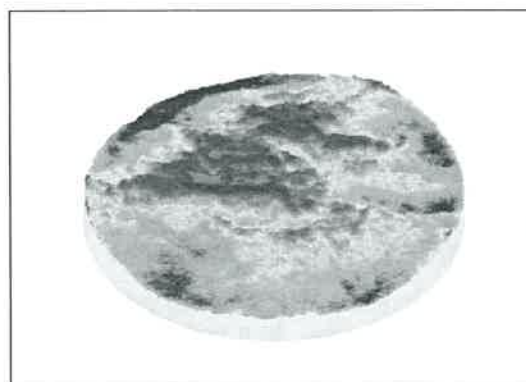


FIGURE 3. 3D surface topography image of a 30-mm diameter SiC standard flatness artefact. The measured S_q is 3.9nm.

PRECISION MACHINING EXAMPLES

Principal uses of large field of view CSI include flatness and waviness measurement of automotive, aerospace, semiconductor and, most recently, 3D additive manufacturing

components [17]. CSI also provides the required data for relational metrology such as parallelism or height between separated surfaces. Here we provide examples of measurements on machined metal parts and assemblies.

A first example in FIGURE 4 is a CSI measurement of a fuel injector component having a ground, unpolished surface finish. The surface roughness is such that there are no continuous fringes visible in the instrument. The full 3D image of FIGURE 5 is readily acquired in a few seconds using CSI and the new objective.



FIGURE 4. Photograph of a 29-mm diameter machined fuel injector component.

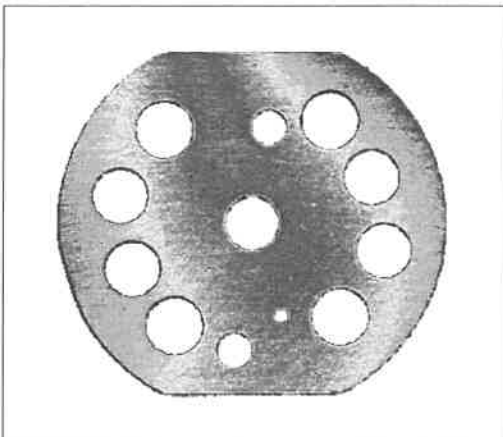


FIGURE 5. 3D CSI image of the part shown in the previous figure. Parameter results: $S_q = 1.106\mu\text{m}$, $S_z = 6.615\mu\text{m}$.

For larger objects, an established technique is the automated stitching of multiple acquisitions with lateral displacements [11]. For stitching measurements, the benefits of the wide field

objective include significantly reduced data acquisition time and improved form metrology as a consequence of fewer stitched fields. As a first example, the pump part of FIGURE 6 is slightly too large for a single data acquisition, but is readily measured with four overlapping fields. The result in FIGURE 7 reveals the top surface form, and the recess of the impeller blades. As a final example, FIGURE 8 shows an assembly of gears in a housing with variable surface textures and reflectivities, successfully measured using stitching of 29 sub-Nyquist CSI image fields into the complete 3D topography shown in FIGURE 9.

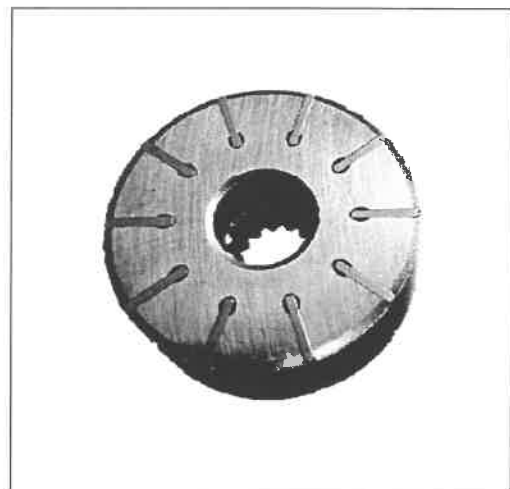


FIGURE 6. Photograph of a 36-mm diameter pump impeller component.

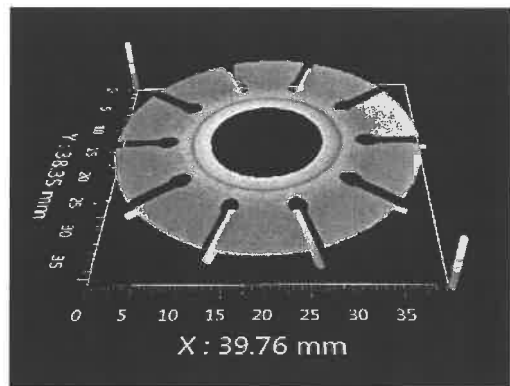


FIGURE 7. 3D CSI image of the part shown in the previous figure.



FIGURE 8. Photograph of a 150-mm diameter transmission pump assembly.

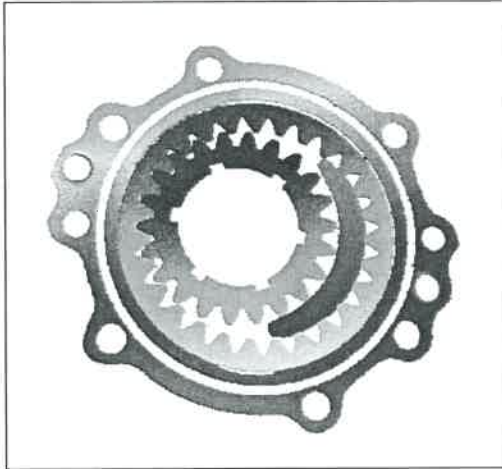


FIGURE 9. 3D image of the part shown in the previous figure. $S_q = 26.7\mu\text{m}$, $S_z = 119.8\mu\text{m}$.

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DEVELOPMENT OF A MINIATURE, MULTICHANNEL, EXTENDED-RANGE FABRY-PEROT FIBER-OPTIC LASER INTERFEROMETER SYSTEM FOR LOW FREQUENCY SI-TRACEABLE DISPLACEMENT MEASUREMENT

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INTRODUCTION

Laser interferometry has become a staple of SI-traceable displacement measurement techniques. With decreasing cost per channel, miniaturization, and a multitude of commercially available off-the-shelf systems, these systems create ever more attractive applications unheard-of a few years ago. Although significant efforts have been made to shrink the size of the optical components necessary for a fully functioning laser interferometer sensor, most currently available commercial sensors have a sensing element on the order of millimeters or more [1].

The goal of the work presented in this paper, which builds on our previous experience with fiber-based laser interferometer systems [2], was to build a directly SI-traceable laser interferometer displacement sensor with a sensor head no bigger than a single optical fiber diameter (0.125 mm). The sensor was designed and built specifically to serve as the primary displacement and force gauge sensor in an SI-traceable precision nanoindentation platform (PNP) [3]. Significant efforts were made to assure that each component not only performed a needed function, but also contained known uncertainties or a specific methodology to ascertain uncertainty values.

SENSOR DESIGN

The sensor is based on a fiber-optic, homodyne, low finesse, single detector Fabry-Perot (FP) cavity that is set up between a polished surface and the cleaved end of an optical fiber (FIGURE 1). When the cavity length changes, a Fabry-Perot laser interferometer system has an output best described by an Airy function, and is typically used only near a quadrature point, *i.e.*, the mid-point between an interference maximum

and minimum. At or near that point, the intensity-displacement relation is assumed to be linear to within a 1 % error for displacements on the order of 100 nm for wavelengths near 1550 nm, but displacements larger than this cannot be tracked.

To overcome this limitation and extend the working range of our sensor, the laser wavelength (nominally 1550 nm) is continuously modulated sinusoidally at a frequency of $f = 1.2$ kHz, which was chosen based on the upper modulation frequency limit of our rapidly tunable laser (RTL). By sinusoidally modulating the wavelength while cavity length, h , is being changed, the signal measured by the detector has a spectrum consisting of responses at the modulation frequency f and its harmonics. The optimal modulation depth depends on the cavity length, and typically is 0.5 nm (peak-to-peak) for cavities on the order of few mm. Taking a closer look at the f and $2f$ signal intensities, I_f and I_{2f} respectively (FIGURE 2), it can be seen that when the cavity is changing they resemble sine

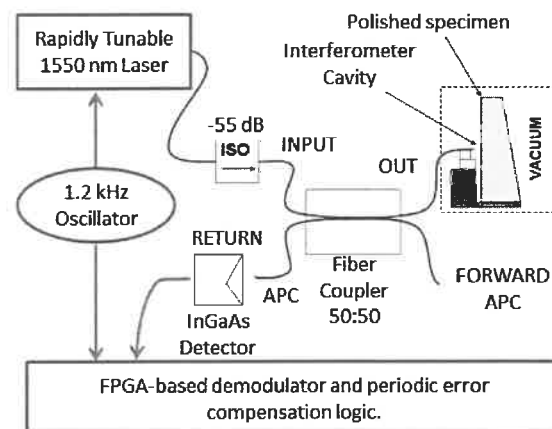


FIGURE 1. Main system components.

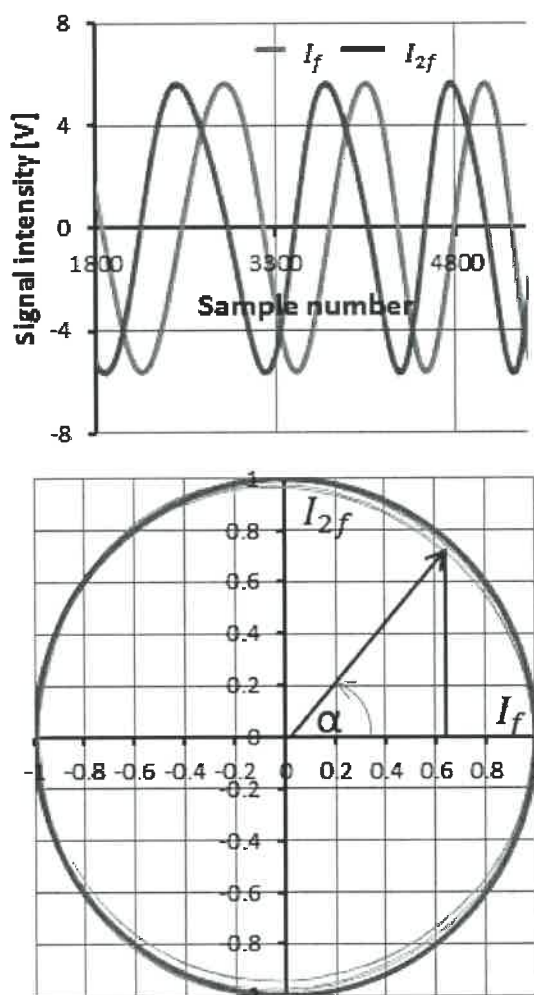


FIGURE 2. (Top) Demodulated signals at f and $2f$ during mirror displacement. (Bottom) f and $2f$ combined into a rotating vector.

and cosine. Those signals can be combined into a rotating vector created between point (0,0) and point (I_f, I_{2f}). Changes in cavity length caused by motion of the polished surface can then be calculated from:

$$\Delta h = \frac{\lambda}{2} \left(\frac{\alpha}{360} + n \right), \quad \alpha = \arctan \left(\frac{I_{2f}}{I_f} \right),$$

where λ is the wavelength, n is the number of times the vector completes a full revolution, and α is the angle (in degrees) formed between the vector and the horizontal axis. This approach extends the working distance of a laser interferometer to well beyond 25 mm, while still maintaining sub-nanometer resolution.

ERROR COMPENSATION STRATEGIES

Compensation for fringe asymmetry

One inherent problem related to using a FP cavity in this extended mode is the inclusion of periodic errors described by Wilkinson *et al.* [4]. We have developed two methods for reducing these errors: one based on a look-up table created during system operation, and another based on post-processing the data using the model developed in Ref. [4].

Experimental approach to error compensation

In this method, a cavity is continuously swept by an open-loop piezoelectric-based nanopositioner capable of a maximum displacement of approximately 35 μm . It is generally known that piezoelectric actuators, if operated in voltage-controlled mode, exhibit nonlinear voltage-displacement characteristics. During a 35 μm cavity sweep with 1550 nm laser wavelength, the rotating vector completes a maximum of 45 full revolutions. Because all these data can be stored and overlapped, they

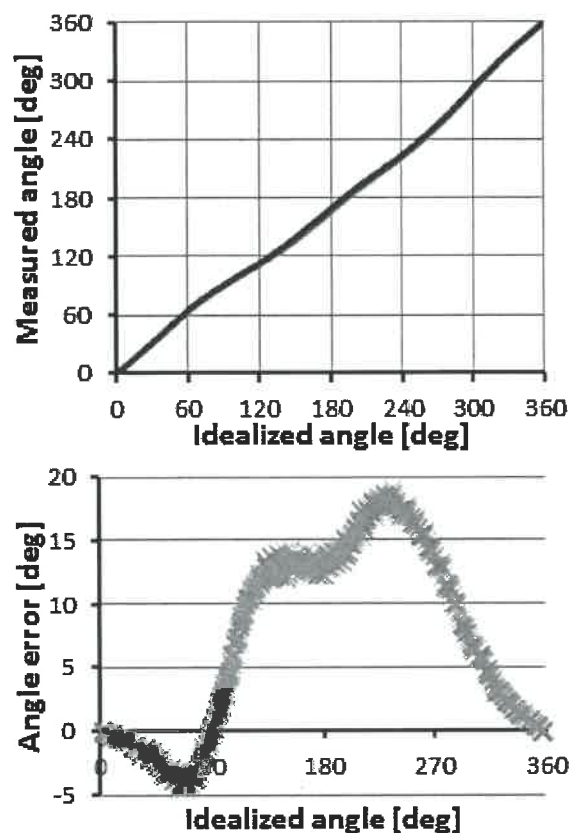


FIGURE 3. Residual periodic error mapped during laser initialization (10 cycles overlapped).

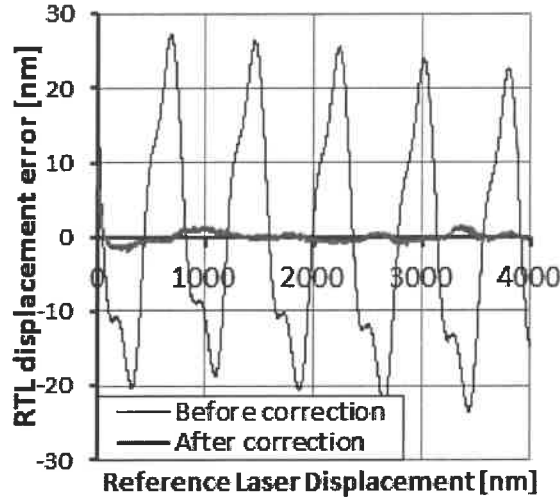


FIGURE 4. Residual periodic error before and after correction, relative to a reference laser.

can be averaged in post-processing to reveal cyclic nonlinearities, as shown in FIGURE 3. In this figure (top), the idealized angle is reconstructed; that is, the angle that the vector should have if no nonlinearities existed, is plotted against the actual measured angle. In the bottom figure, multiple overlapped cycles are presented with angular errors extracted. A bicubic spline function is then used to fit the data to average all overlapping cycles into one monotonic data set. These data form a base for creating a 7200 point lookup table with linear interpolation used in between points.

By using this method alone, it was possible to reduce the amplitude of the periodic errors from approximately 50 nm down to the single-nanometer level (FIGURE 4).

Analytical approach to error compensation

This section provides a brief overview of an ongoing study to evaluate a theoretical model of this interferometer implementation. Assuming an approximately Gaussian beam profile, Nemoto and Nakimoto [5] produced a theoretical model for predicting beam coupling between two fibers. Wilkinson and Pratt [4] adapted this to model field reflectance of the fiber and mirror interferometer. In their paper, they develop the following equation to model the coupling of each successive reflection of the cavity. In this model the coupling can be expressed in terms of the n^{th} reflection (this should not be confused with the refractive index of the cavity n_c). Hence the coupling C_n for the n^{th} reflection is given by [4]

$$C_n = \Delta e^{i\beta_n} e^{i\Theta_n}$$

$$\Delta_n = \frac{1}{\sqrt{1+n^2\bar{z}_m^2}} e^{-\left[\frac{kz_R(1+5n^2\bar{z}_m^2)}{1+n^2\bar{z}_m^2} n^2\theta_m^2\right]}$$

$$\Theta_n = -\tan^{-1}(n\bar{z}_m) + \frac{kz_R(3n\bar{z}_m - n^3\bar{z}_m^3)}{1+n^2\bar{z}_m^2} n^2\theta_m^2$$

$$\beta = 2nkz$$

where k is the wave vector in the cavity ($= 2\pi n_c / \lambda$), z_R is the Rayleigh length ($= ks^2$), s is the spot radius, λ is the vacuum wavelength of the illumination source, $\bar{z} = z / z_R$ is the normalized separation of the cavity, $\bar{x} = x / z_R$ is the normalized lateral (or transverse) shift of the reflecting beam and θ_m is the angular misalignment between the two surfaces of the cavity. Summing the coupling of the beam into the fiber for each reflection, the resultant complex value will provide the field reflectance of the cavity.

Having calculated the coupling coefficient for each reflection of the beam, the total field reflectivity, r , of the cavity is computed from the complex sum of reflections [4]:

$$r = r_o C_o + \frac{t_o^2}{r_o} \sum_{n=1}^{\infty} (r_m r_o)^n C_n e^{-i\pi}$$

where r_m and r_o represent the field reflectivity of the mirror and fiber end-face and t_o the transmitted field. The net result of this analysis is a complex number representing the magnitude and phase of the Fabry-Perot mirror. Squaring the field reflectivity produces the intensity reflectance corresponding to the voltage output of the detector.

In order to model experimental f and $2f$ responses like those shown in FIGURE 2, the first and second modulation harmonics are calculated and compared with the real and imaginary components of the cavity with a glass reflecting surface, FIGURE 5. From this model, a cavity misalignment of 47.7 mrad, minimum separation of 2.0056 mm and field reflectivity of 0.2011 were derived from the fit coefficients.

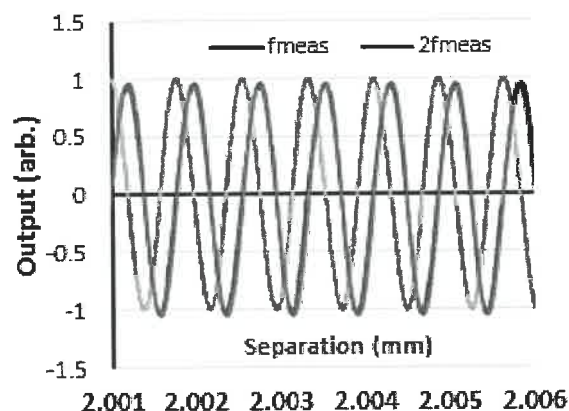


FIGURE 5: Plots of the first two harmonics of the modulation signals from the interferometer with fits of the real (green) and imaginary (purple) response of the theoretical model overlaid.

From the initial tests, the theoretical model above deviates from reflectance measurement of the FP cavity at around the 1% level. While further analysis may reduce these deviations, it is likely that the discrepancies are due to differences between the physics of the model and that of the experimental apparatus and measurement methods. Possible error sources include: non sinusoidal time modulation of the source beam, the normalization methodology used to extract phase from the quadrature signal, the non-sinusoidal nature of the FP cavity reflectance, errors with the assumed Gaussian beam profile, and errors associated with the assumption that the spot size from which the Rayleigh length is calculated is the same as the fiber diameter.

It is mainly because of these uncertainties that a more robust approach based on renormalization and look-up tables is being pursued, an approach also adopted in [1].

Amplitude normalization technique and offset removal

Other sources of periodic errors include differences in f and $2f$ amplitudes, as well as minute DC voltage offsets present in signals. Since basic trigonometry is used to compute the angle from f and $2f$ intensities, it is assumed that their corresponding amplitudes are equal, and the rotating vector forms a perfect circle centered at the point (0,0). Any deviation from that creates periodic errors.

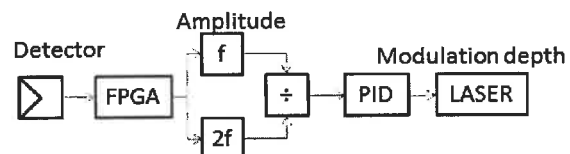


FIGURE 6. Mechanism of reducing periodic errors due to f and $2f$ amplitude difference.

Amplitudes of both f and $2f$ depend strongly on laser wavelength modulation depth, the optimal amplitude of which is dependent on laser cavity length: the longer the cavity, the smaller the modulation depth required.

In order to maintain a constant ratio of f and $2f$ amplitudes, a closed-loop proportional-integral-derivative (PID) system was incorporated; it is presented schematically in FIGURE 6. The signal from a detector is separated into f and $2f$ components using a lock-in amplifier implemented with field-programmable-gate-array (FPGA) methods. Amplitudes and residual offsets of the f and $2f$ signals are then measured individually for every interference cycle. Offsets are then removed by subtracting the measured values from corresponding f and $2f$ signals. The amplitude ratio of f and $2f$ is sent to the PID controller, which changes wavelength modulation depth to keep the ratio at unity.

ABSOLUTE CAVITY LENGTH MEASUREMENT

Because the laser wavelength can be easily changed by applying an input voltage to the laser control module, the absolute cavity length can be measured by sweeping wavelength and counting the number of times the interferometer signal reaches its peak. Knowing the wavelength at the first maximum (λ_1) and last (λ_2) and the number of interference cycles (m) in between, the absolute cavity length (h) can be computed from the following equation:

$$h = \frac{m}{2} \left(\frac{\lambda_1 \lambda_2}{\lambda_2 - \lambda_1} \right).$$

The biggest contributor to uncertainty in this cavity length determination method is how accurately wavelengths at interference maxima can be determined.

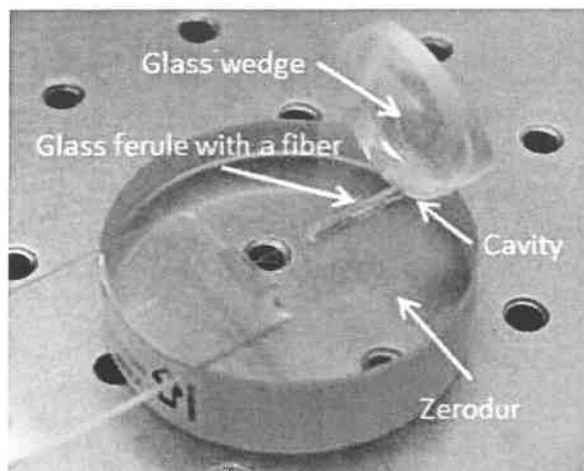


FIGURE 7. Stable cavity assembly for laser stability test.

SENSOR PERFORMANCE

Sensor stability and noise

There are various methods of stabilizing the laser wavelength in interferometry applications. Many systems have been developed using a reference gas absorption cell or a reference Fabry-Perot cavity. In this work, laser stability is achieved by measuring wavelength with an optical spectrum analyzer (OSA). The OSA continuously sends wavelength and modulation depth information through an RS-232 interface to a real-time control module used as a PID controller. This scheme allows low-picometer wavelength stability while maintaining the flexibility to choose the operating wavelength.

In order to test how stable the displacement measurement is in a fully operational instrument deployed in its working environment, a test setup was built (see FIGURE 7). A glass ferrule holding an optical fiber was mounted with a dab

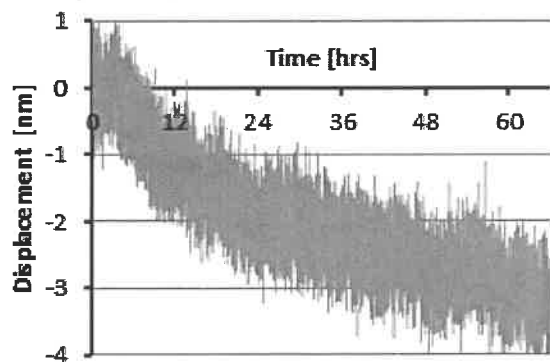


FIGURE 8. Laser stability test over 66 hours (data acquired at a rate of 10 samples per minute).

of glue to a glass substrate having a very low coefficient of thermal expansion. An optical cavity was formed between a cleaved fiber end and an optically polished glass wedge attached to the substrate. Care was taken to assure that the metrology loop was as short as feasible. The assembly was then placed in a temperature-controlled chamber capable of maintaining a temperature setpoint to approximately 0.01 °C.

Data obtained during the logging of displacement are shown in FIGURE 8. Data were recorded over 66 h at a rate of 10 samples per minute. The average stability of the sensor during this experiment was measured to be approximately 1 nm per 24 h. The exponentially decaying drift is likely caused by handling the assembly prior to recording data. Higher-frequency variations in displacement are caused primarily by the dynamics of the environmental chamber PID temperature control system.

The noise performance of the system at higher frequencies can be seen in FIGURE 9. Here data were recorded at the rate of 74 Hz. Data obtained follows a normal distribution with 6σ of 0.914 nm and noise amplitude $0.106 \text{ nm}/\sqrt{\text{Hz}}$.

Maximum range for a mirror-polished target

The maximum working range of our system, which uses a source laser power of approximately 5 mW, was determined by moving the cleaved fiber end away from target until accurate nanometer level displacement measurement was no longer possible. The maximum distance was determined to be approximately 25 mm, which is more than adequate for the current application.

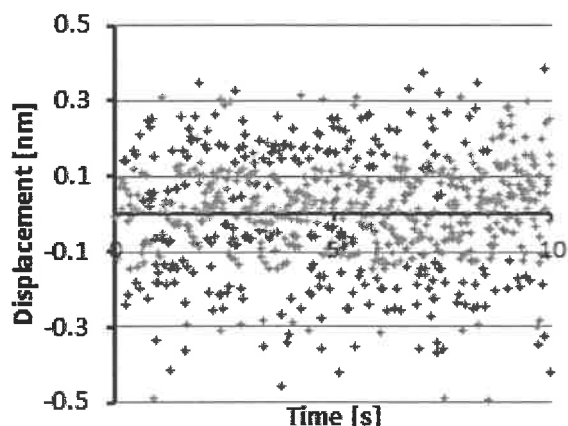


FIGURE 9. Laser system displacement noise measurement.

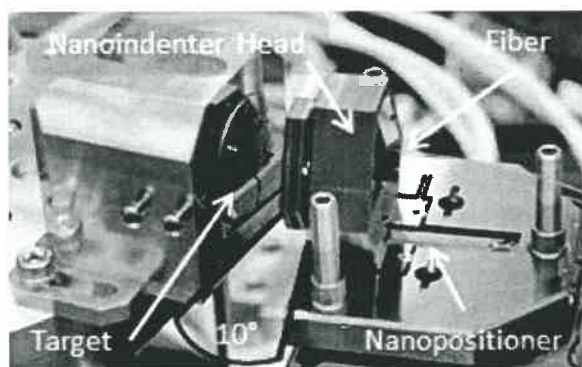


FIGURE 10. Fiber-target misalignment.

Angular misalignment and target roughness

One major advantage of using a bare cleaved fiber is that careful fiber-target alignment using focusing lenses is not required. The maximum possible misalignment at which the interferometer would still work was $\pm 10^\circ$, when tested on an optically polished aluminum target. The setup and misalignment can be seen in FIGURE 10. This feature greatly reduces set-up time and precision required for the fiber-target pair during assembly. Although the system can function with high degrees of misalignment, it is not advised to operate at those limits without careful consideration of the beam path to reduce unwanted uncertainty.

Typical laser interferometer systems require an optically polished target surface to operate properly. To test how our system behaves when the target surface is of significantly poorer optical quality, we used a set of targets made from common nanoindentation specimens. They are presented in FIGURE 11. In order they are: 1) a cured epoxy resin, 2) poly (methyl methacrylate), 3) a polished polycrystalline ceramic, 4) a silicon wafer, and 5) high-density polyethylene.

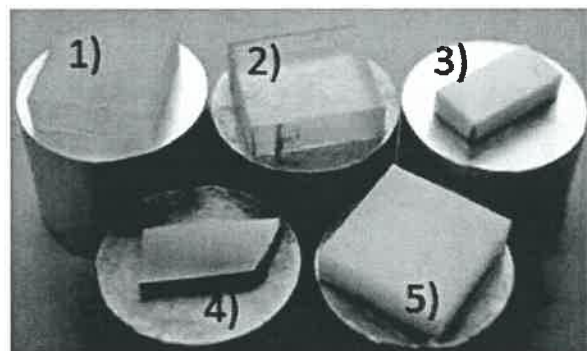


FIGURE 11. A variety of materials that all created functional Fabry-Perot cavities.

There was no distinguishable difference between any of those samples and a polished Al mirror in the performance of the system. The interferometer system was stable and the noise level remained at the reference level.

CONCLUSION

This paper presents the development of a miniature, multichannel, extended Fabry-Perot fiber-optic laser interferometer system designed for a precision SI-traceable nanoindentation application. The system meets all requirements, achieving sub-nanometer noise levels, high stability, immunity to minute fiber-target misalignments and a surprising level of immunity to surface roughness. Periodic errors can be successfully reduced using the methods described.

DISCLAIMER

Commercial equipment and materials may be identified in order to adequately specify certain procedures. In no case does such identification imply recommendation or endorsement by the National Institute of Standards and Technology, nor does it imply that the materials or equipment identified are necessarily the best available for the purpose.

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A NOVEL HETERODYNE GRATING INTERFEROMETER SYSTEM FOR IN-PLANE AND OUT-OF-PLANE DISPLACEMENT MEASUREMENT WITH NANOMETER RESOLUTION

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INTRODUCTION

In the past, the multi-axis heterodyne laser interferometer system is the main tool for the six-degree-of-freedom (DOF) displacement measurement with nanometer-level accuracy of the wafer stage in a lithography scanner [1-2]. However, as the semiconductor process is scaling down continuously, the accuracy enhancement of this system is becoming more and more difficult due to its high environmental sensitivity. Hence, an environmental robustness grating encoder system is more favorable to meet the ever-increasing measurement requirement of wafer stage in non-vacuum environment [1-2]. In addition, a special grating encoder which can simultaneously measure a long stroke with hundreds of millimeters along in-plane direction and a short stroke with hundreds of micrometers along out-of-plane direction with nanometer-level accuracy is one of the key components for six-DOF displacement measurement grating encoder system of wafer stages [3].

Former work of interferometry based homodyne grating encoders which could simultaneously measure multi-dimensional displacement has been demonstrated in some reports [4-7]. Nevertheless, these proposed encoders have two disadvantages by the reason of adopting homodyne detection scheme, which limit its application in wafer stages. For the first, direct-current (DC) interference signal of these homodyne grating encoders suffers from susceptibility to random disturbance such as laser power variation, geometry alignment and electronic noise, which have a negative effect on the accuracy of the encoders [4-6]. For the second, the optical setup of these homodyne grating encoders usually have an optical phase shifting structure for homodyne quadrature detection, which make the optical setup large and complicated [4-6].

This paper describes a novel heterodyne grating interferometer system which can simultaneously measure a long stroke with hundreds of millimeters in the in-plane direction and a short stroke with hundreds of micrometers in the out-of-plane direction. The displacement resolution is 1.63nm along the x-direction and 0.75nm along the z-direction. Due to the utilization of the heterodyne detection scheme, higher AC heterodyne signal anti-disturbance capability, less optical components and more compact optical setup can be achieved as compared to the homodyne counterpart. First accuracy test results show that the measurement standard deviation is 6.37nm in the in-plane direction, and is 3.69nm in the out-of-plane direction.

We aim to improve the displacement measurement accuracy of the wafer stage in the lithography scanner in the future, by means of this novel environmental robustness heterodyne grating interferometer system for simultaneous in-plane and out-of-plane displacement measurement with nanometer accuracy.

CONSTRUCTION OF NOVEL HETERODYNE GRATING INTERFEROMETER SYSTEM

Structure of Novel Heterodyne Grating Interferometer System

Figure 1 shows the schematic diagram of the proposed two-dimensional heterodyne grating interferometer system. The grating interferometer system is composed of a dual-frequency laser head, a reflective-type scale grating, a read head, two receivers and a two-axis phase meter. The read head consists of a polarizing beam splitter, a reflective-type reference grating, four mirrors and two fiber pickups with a built-in polarizer.

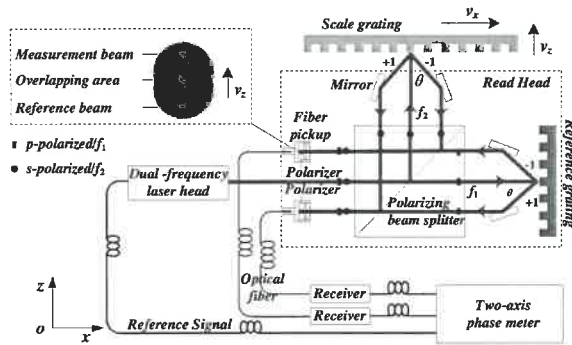


FIGURE 1. The Proposed novel heterodyne grating interferometer system diagram and measurement principle.

A dual-frequency laser head radiates a beam which contains two frequency components with unique linear polarization (one frequency component is f_1 with p-polarization, and the other is f_2 with s-polarization). The beam is split into a measurement and reference optical path at a polarizing-beam-splitter (PBS). The frequency component f_1 with p-polarization is used as the measurement beam and diffracts from the moving scale grating back to the PBS. The other frequency component f_2 with s-polarization diffracts from a fixed reference grating back to the PBS. At the PBS, the positive and negative first-order diffracted beams from the scale grating respectively combine with the corresponding diffracted beams from the reference grating. The combined beams form two optical heterodyne signals at the two fiber pickups, then detected by two receivers, respectively. Two electronic measurement signals from the receivers and an electronic reference signal from laser head, separately access to the two-axis phase meter, finally, two-dimensional displacements are read out by the two-axis phase meter with heterodyne phase detection scheme.

Displacement Measurement Principle of Heterodyne Grating Interferometer System

As the Figure 1 (upper right) illustrated, if there are relative displacements Δx in x direction and Δz in z direction between grating interferometer and scale grating, following phase changes will be detected by two-axis phase meter:

$$\begin{cases} \Phi_1 = -2\pi \left[\frac{\Delta x}{\Lambda} + (1 + \cos \theta) \frac{\Delta z}{\lambda} \right] \\ \Phi_2 = -2\pi \left[-\frac{\Delta x}{\Lambda} + (1 + \cos \theta) \frac{\Delta z}{\lambda} \right] \end{cases}, \theta = \arcsin \left(\frac{\lambda}{\Lambda} \right)$$

Where θ is the +1/-1 diffraction angle when the beam vertically incidents the scale grating, Λ is the grating pitch, and λ is the wavelength of the laser.

The electric field vector of the incident beam with the orthogonal polarization is defined as:

$$E = \begin{bmatrix} E_0 \exp[-i(2\pi f_1 t + \phi_0)] \\ E_0 \exp[-i(2\pi f_2 t + \phi_0)] \end{bmatrix}$$

After passing through the polarized optical parts, the electric field vector of four diffraction beams, i.e., E_r and E_m , can be written as:

$$\begin{cases} E_{r(+1)} = J_{P(45^\circ)} J_{PBT} J_{RG(+1)} J_{PBT} E \\ E_{r(-1)} = J_{P(45^\circ)} J_{PBT} J_{RG(-1)} J_{PBT} E \\ E_{m(+1)} = J_{P(45^\circ)} J_{PBR} J_{MG(+1)} J_{PBR} E \\ E_{m(-1)} = J_{P(45^\circ)} J_{PBR} J_{MG(-1)} J_{PBR} E \end{cases}$$

Where $\mathbf{J}$ is the Jones matrix and is explicitly pointed out in the Table 1.

TABLE 1. The Jones Matrix of the optics in the heterodyne grating interferometer system.

Optics	Jones Matrix
Polarizer	$J_{P(45^\circ)} = \frac{1}{2} \begin{bmatrix} 1 & 1 \\ 1 & 1 \end{bmatrix}$
PBS	$J_{PBT} = \begin{bmatrix} 1 & 0 \\ 0 & 0 \end{bmatrix}, J_{PBR} = \begin{bmatrix} 0 & 0 \\ 0 & 1 \end{bmatrix}$
Scale Grating	$\begin{cases} J_{MG(+1)} = \begin{bmatrix} \sqrt{\eta_p} e^{i\Phi_1} & 0 \\ 0 & \sqrt{\eta_s} e^{i\Phi_1} \end{bmatrix} \\ J_{MG(-1)} = \begin{bmatrix} \sqrt{\eta_p} e^{i\Phi_2} & 0 \\ 0 & \sqrt{\eta_s} e^{i\Phi_2} \end{bmatrix} \end{cases}$
Reference Grating	$J_{RG(+1)} = J_{RG(-1)} = \begin{bmatrix} \sqrt{\eta_p} & 0 \\ 0 & \sqrt{\eta_s} \end{bmatrix}$

Where η_s is the diffraction efficient of p-polarization beam, and η_p is the diffraction efficient of s-polarization beam for the scale and reference grating.

Finally, the electric field vector of the optical heterodyne signals at the receivers can be written as:

$$\begin{cases} E_{M1} = E_{m(+1)} + E_{r(+1)} \\ E_{M2} = E_{m(-1)} + E_{r(-1)} \end{cases}$$

Therefore, the heterodyne signals respectively detected by two receivers are:

$$\begin{cases} I_{M1} \propto E_{M1} \overline{E_{M1}} \propto \cos[2\pi(f_1 - f_2)t - \Phi_1] \\ I_{M2} \propto E_{M2} \overline{E_{M2}} \propto \cos[2\pi(f_1 - f_2)t - \Phi_2] \end{cases}$$

As Φ_1 and Φ_2 could be read out by the two-axis phase meter, the displacement values of Δx and Δz can be resolved as:

$$\begin{cases} \Delta x = \frac{\lambda}{4\pi}(\Phi_1 - \Phi_2) = \frac{\lambda(k_1 - k_2)}{2N} \\ \Delta z = \frac{\lambda}{4\pi(1 + \cos\theta)}(\Phi_1 + \Phi_2) = \frac{\lambda(k_1 + k_2)}{2(1 + \cos\theta)N} \end{cases}$$

Where k_1 and k_2 are the counts of phase meter, N is the subdivision value of the phase meter, and the phase resolution of the phase meter is $2\pi/N$.

In addition, as shown in Figure 1 (upper left), the overlapping area of measurement and reference beam will shrink when the scale grating moves along the out-of-plane direction in a small range. Nevertheless, only overlapping area generates the measurement signal which contains the displacement information. Thus, the out-of-plane measurement stroke is restricted to get enough overlapping area.

Experiment Setup of the Heterodyne Grating Interferometer System

We establish an experiment setup of the heterodyne grating interferometer system, as shown in Figure 2. The experiment setup employs two identical holography gratings with a pitch of 833.3nm—one is the moving scale grating and the other is the fixed reference grating. The Agilent 5517C laser head is used to provide dual-frequency orthogonal polarization laser with 632.8nm wavelength. The frequency difference of the dual-frequency is 2.7MHz and the diameter of the beam is 6mm. The setup also employs Agilent E1709A as the receiver and Agilent 10898A as the two-axis phase meter with a $2\pi/256$ phase resolution. For Agilent E1709A receivers, the overlapping size cannot be lower than 80% of the beam spot size, so the

out-of-plane measurement stroke is no more than 1.2mm. And for this experiment setup, the displacement resolution is 1.63nm along the x-direction and 0.75nm along the z-direction.

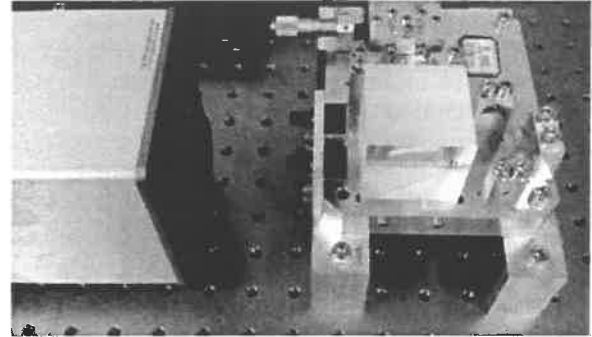


FIGURE 2. The Photograph of the novel heterodyne grating interferometer.

COMPARISON EXPERIMENT BETWEEN NOVEL HETERODYNE GRATING INTERFEROMETER SYSTEM AND HETERODYNE LASER INTERFEROMETER SYSTEM

Experiment Setup

To verify the measurement result of the heterodyne grating interferometer, another Agilent measurement system with two plane mirror interferometers is built for comparison. The resolution of Agilent measurement system is 0.62nm. Figure 3 shows the photograph of the comparative measurement system setup.

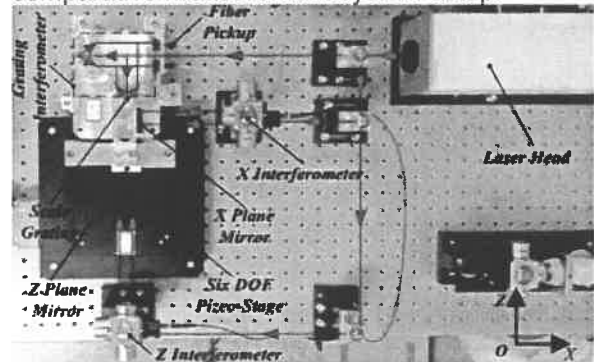


FIGURE 3. The Photograph of novel heterodyne grating interferometer and comparative system experiment setup.

In this comparison system setup, the scale grating and the plane mirrors are mounted on the moving part of a PI six-axis piezo-stage. The read head of the grating interferometer, the laser interferometers and the stationary part of the piezo-stage are fixed on the optical bench. The Agilent 5517C laser head is used to provide dual-frequency orthogonal polarization laser for

the grating interferometer and plane-mirror interferometers. The optical path of the comparison system is also illustrated in Figure 3.

Experiment Result

The previously-mentioned piezo-stage is operated to move in sinusoidal trajectories, and the magnitudes are $1\mu\text{m}$ along the x-direction and $0.1\mu\text{m}$ along the z-direction, respectively. The experiment results are separately illustrated in Figure 4 and Figure 5.

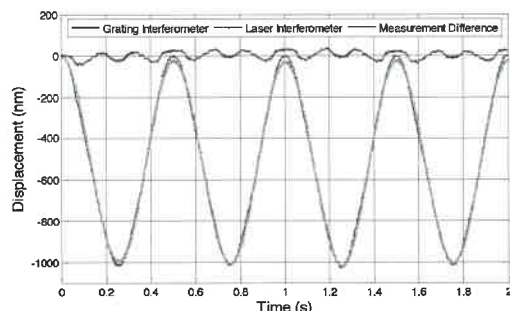


FIGURE 4. The displacement measuring curves and the measurement discrepancy along x-direction.

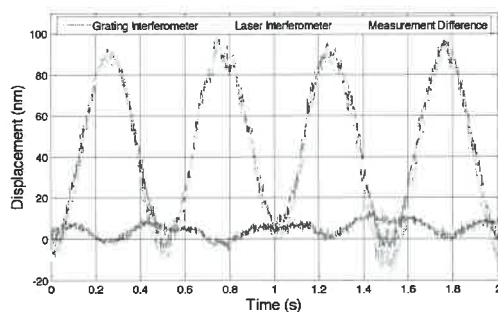


FIGURE 5. The displacement measuring curves and the measurement discrepancy along z-direction.

A primarily accuracy test of the novel heterodyne grating interferometer system is also done based on the comparative experiment system. The piezo-stage is operated to move about $1\mu\text{m}$ in x direction, and 100nm in z direction for 20 times. The measurement data of 20 times is recorded and processed, and the mean and stand deviation of the measurement data are illustrated in Table 2.

TABLE 2. The accuracy test result of the novel heterodyne grating interferometer system and laser interferometer for comparison.

		Mean	Stand Deviation
Grating Interferometer	Xg (nm)	-995.09	6.37
	Zg (nm)	89.43	3.69
Laser Interferometer	Xi (nm)	-963.22	5.56
	Zi (nm)	95.70	4.56

The measurement results illustrated in the figures and table between grating interferometer and interferometer show good consistency with each other, but there are little values of the measurement discrepancy are (31.87nm, 0.81nm) in x-direction and (6.27nm, 0.87nm) in z-direction, respectively. We consider that the discrepancy may result from the error of grating interferometer and Abbe error of the comparative system. It is possible to further improve the difference by optimizing the design of the two-dimensional grating interferometer and minimizing the alignment errors. Nevertheless, the goal to verify the principle of the heterodyne grating interferometer is achieved by this experiment.

RESULT

A novel heterodyne grating interferometer system which can simultaneously measure a long stroke with hundreds of millimeters in the in-plane direction and a short stroke with hundreds of micrometers in the out-of-plane direction is proposed. The displacement resolution is 1.63nm along the x-direction and 0.75nm along the z-direction. The primarily accuracy test results show that the measurement standard deviation is 6.37nm in the in-plane direction, and is 3.69nm in the out-of-plane direction.

FUTURE WORK

Further developments of this type of grating interferometer will focus on three points for the application in wafer stage:

- The resolution improvement based on optical frequency multiplication method and electronic subdivision technology;

- The accuracy enhancement based on error analysis and calibration method;
- The optical fiber transmission technology for orthogonal polarization dual-frequency laser's transmission from laser head to grating interferometer.

In the future, this type of grating interferometer system together with large scale planar grating will be used for the six degree-of-freedom displacement measurement with nanometer accuracy in wafer stage.

ACKNOWLEDGEMENTS

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PHASE COMPENSATION FOR DYNAMIC DOPPLER FREQUENCY SHIFTS

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INSTRUCTIONS

Displacement measuring interferometry is a widely used technique for displacement metrology. There are two types of displacement interferometers: homodyne and heterodyne. In all cases, the metrology principle is the same: target position changes are recorded as a Doppler frequency shift in the optical frequency, which manifests as a measured phase change. The target displacement is determined by the measured phase changes. As the velocity of the measurement target changes, the instantaneous frequency of the detected signal changes. The relationship between the Doppler frequency shift and the velocity of target is

$$f_D = \frac{1}{2\pi} \frac{d\phi}{dt} = \frac{Nn}{\lambda} \frac{dx}{dt}, \quad (1)$$

where ϕ is the phase difference between the reference and measurement signals, N is the interferometer fold factor (two for the interferometer used in this work), n is the refractive index along the optical path difference, λ is the nominal wavelength of the laser light, and x is the displacement of the target.

For heterodyne interferometry, the two optical frequencies from the laser source provide a quasi-static heterodyne frequency, f_s . The combination of a nominal heterodyne frequency, f_s , with a moving target inducing a Doppler frequency shift, f_D , means the frequency of the measurement signal is $f_s \pm f_D$. As the target velocity changes, the instantaneous frequency of the measurement signal changes, leading to a different local phase response as a function of detection frequency due to filtering in the measurement [1]. This imparts a distortion in the measured phase change, leading to errors when there are rapid position changes and non-constant velocity motions of the target.

SIGNAL PROCESSING

Prior to being processed, two optical signals (measurement signal and reference signal) are detected and converted to voltage level representations of interference amplitude. The measurement signal has an instantaneous frequency of $f_s \pm f_D$ and its phase is measured against the reference signal at f_s . Typically, the analog signal is processed prior to converting it to a digital signal to maximize the signal and quality in the analog-to-digital converter (ADC). Figure 1a shows the analog processing used in this work, which is designed for a 70 kHz heterodyne frequency, with a maximum Doppler frequency shift of 20 kHz. Figure 1b shows the modeled phase response for this system.

In this work, a single-bin discrete Fourier transform (SBDFT) [2] is used as phase measurement method. Figure 2 shows an example of a SBDFT technique and its LPFs' Bode plot that we used as part of this research with our heterodyne interferometer. The input signals are reference and measurement signals, which have experienced the analog processing; the output signal is the phase difference between two input signals. The detail mechanism can be found in, e.g., Ref. [2, 3]. Once the phase is known, the displacement can be calculated from the phase difference from the start of a measurement, scaled with the interferometer fold factor, N , refractive index, n , and wavelength, λ .

DYNAMIC DOPPLER FREQUENCY SHIFTS

When the desired signal passes through a filter with varied frequency, a frequency-dependent phase delay will be introduced due to the non-constant phase response of the filters. This problem will occur when the target, which is measured by the interferometer, moves with a varied velocity. However, this phenomenon widely exists in phase measurement. Because filters are used to remove noise or demodulate, signals always have a non-constant phase delays

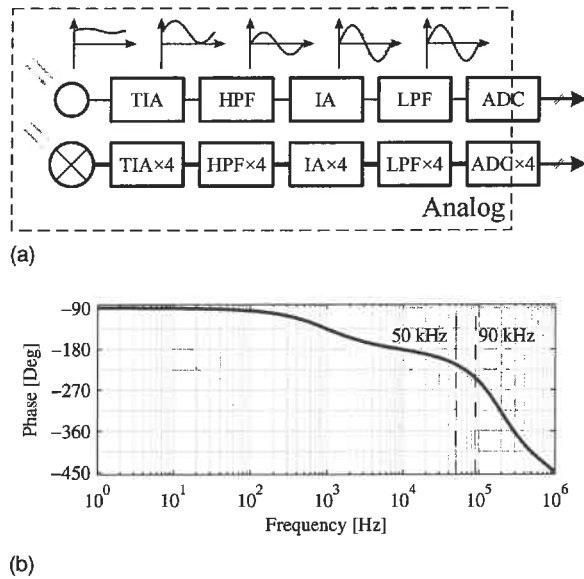


FIGURE 1. The schematic of the analog processing system (a) and its modeled phase response (b). The system employs several op-amp circuits: HPFs, LPFs, TIAs, IAs, to process the analog signal. However, it delays the phase of signal by a function of frequency (b), which is the superimposition of those circuits' phase responses. In the band of 70 ± 20 kHz, the phase difference caused by this is about 30° or 27 nm.

even over narrow pass bands, and the filters are irreplaceable in most phase measurement algorithms. This delay can be a critical phase measurement error in any application where the velocity is varied continuously, such as in semiconductor equipment and inspection, dynamic position calibration, and pick-and-place machines. Thus, compensating the phase delay due to non-constant phase response of filters is crucial to accurate phase measurements, especially when there are high velocities and accelerations in the system. Additionally, there is a recent push to have a calibration standard for dynamic positioning errors, which must account for this error [4, 5, 6, 7].

The phase measurement in heterodyne displacement interferometry systems consists of two main sources of dynamic frequency shift error: 1) filtering in the high frequency detection regime, and 2) filtering after the phase has been converted to a quasi-DC level signal. The high frequency regime consists of any processing that would cause a frequency-dependent phase delay prior to the signal being multiplied by the signals from the VCO, which is the analog processing

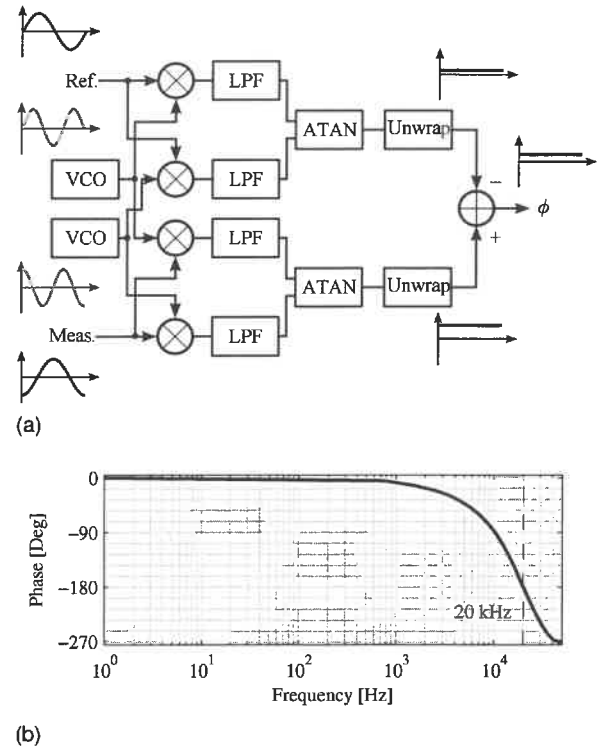


FIGURE 2. The schematic of the digital processing (a) which is based SBDFT and implemented in FPGA in this design; and the phase response of the digital low-pass filters (b). Within its pass band 0 to 20 kHz, up to 180° phase delay or 160 nm displacement error could be introduced.

system in this work (Figure 1). In the high frequency regime, HPFs and LPFs are typically used to remove the DC component and high frequency noise of the signals, contributing to the phase delay. Similarly, the low frequency regime is for phase shifts induced after the multiplication with the VCO, which is the digital processing system in this work (Figure 2). The filters in low frequency regime are typically used to demodulate the phase.

Assume the input measurement and reference signals are

$$u_m(t) = U_m \sin(2\pi f_s t + \phi_m) + U_{DC} + U_{Noise}, \quad (2)$$

$$u_r(t) = U_r \sin(2\pi f_s t + \phi_r) + U_{DC} + U_{Noise}, \quad (3)$$

where U_m and U_r are the amplitudes of the measurement and reference signals, f_s is the nominal heterodyne frequency of the laser source, ϕ_m and ϕ_r are the phases of the two signals (could be functions of time), and U_{DC} and U_{Noise} are the DC offset and noise, which may

differ in the two signals.

After processing in the high frequency regime and converting to digital for mixing and low frequency processing [8], the relative phase difference $\Delta\phi'$ between the measurement and reference signals is

$$\begin{aligned}\Delta\phi' &= \phi'_m - \phi'_r \\ &= \phi_m - \phi_r \\ &\quad + \left(\phi_{h,m}(f_m) + \phi_{l,m}(f_{d,m}) - \phi_{h,r}(f_r) - \phi_{l,r}(f_{d,r}) \right).\end{aligned}\quad (4)$$

where $\phi_h()$ and $\phi_l()$ are the high frequency and low frequency regime phase responses (Figure 1b and Figure 2b), f_m and f_r are the instantaneous frequency of the two signals, and $f_{d,m}$ and $f_{d,r}$ are the Doppler frequency shifts.

The ideal result should be $\phi_m - \phi_r$, however, the extra phase terms, $\phi_{h,m}(f_m)$, $\phi_{l,m}(f_{d,m})$, $\phi_{h,r}(f_r)$, and $\phi_{l,r}(f_{d,r})$, contribute some error and are a function of the Doppler frequency shift. For quasi-static measurements, these added phase terms can be largely ignored. For dynamic measurements, these added phase errors can contribute significant error and must be considered.

Simulations were performed to illustrate how these extra phase terms or phase delays impact the final result when target is moving with constant and non-constant velocity profiles. As shown in Figure 3, filters introduce constant phase delay when target moving in constant velocity, which does not cause an error in the measured phase value because the relative displacement between any two time points is the same in theory and in practice. However, when the target is moving with non-constant velocity, the phase delay and the relative displacement change along with Doppler frequency shift (velocity of the target), which is undesirable in practice. This issue can present a pervasive problem because even under quasi-static conditions, the target still has a nominal Doppler frequency shift value. To improve the dynamic performance of the phasemeter under dynamic conditions, we present potential compensation methods to significantly reduce this error.

COMPENSATION METHODOLOGIES

Conceptually, the solution to compensate this phase error is to subtract the phase delay from the measurement result. From (4), the four varied

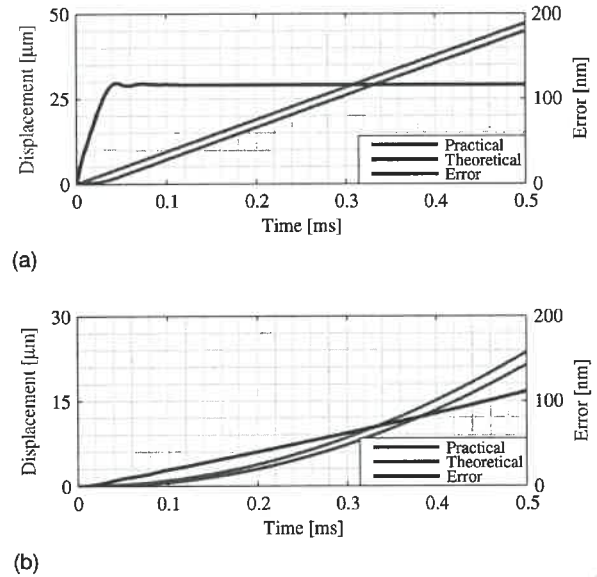


FIGURE 3. The displacements and errors of the target moving with constant (a) and non-constant (b) velocity profiles. The phase (displacement) delay is a function of Doppler frequency shift (velocity of the target). In (a), the displacement error is constant, which does not cause an error in the measured phase value because the relative displacement between any two time points is the same in theory and in practice. In (b), the phase delay and the relative displacement change along with Doppler frequency shift (velocity of the target), which is undesirable in practice.

terms contribute to the phase delay. The high- and low-frequency regime phase responses $\phi_h()$ and $\phi_l()$ must be known prior to the measurement, and frequencies f_m , $f_{d,m}$, f_r , and $f_{d,r}$ should be tracked in real time. To compensate for those errors in real time, the four phase terms $\phi_{h,m}(f_m)$, $\phi_{l,m}(f_{d,m})$, $\phi_{l,r}(f_r)$ and $\phi_{l,r}(f_{d,r})$ must be determined in real time, too. The schematic of overall phase compensation module is shown in Figure 4, which includes an instantaneous frequency monitoring part and a phase delay computation part. The following sections detail two different methods for monitoring the instantaneous frequency and two methods for computation phase delay.

Phase Response Measurement

The preliminary step in the process is to determine the phase responses of the system, specifically from analog and digital filtering. Although the filter parameters are nominally known, variations in component tolerances (analog) and difference from fixed point calculations (digital)

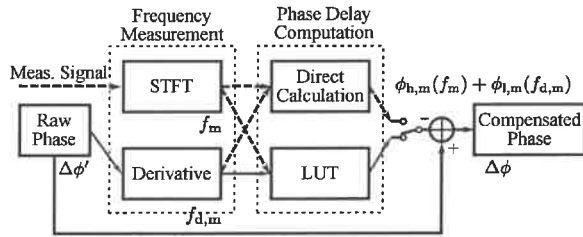


FIGURE 4. Phase compensation module for measurement signal, the same as that for reference signal. It consists of two sections, the frequency measurement section and phase delay computation section. Each section has two potential methods to implement. In this design, the path with red arrow was followed. First, it applies derivative on raw phase to extract instantaneous Doppler frequency shift. Then, it looks up the corresponding phase in LUT. Last, it subtracts the phase delay from raw phase, and gets compensated phase.

cause deviations from the desired performance. For analog components, the performance of each amplifier chip, capacitive coupling in the board and near-field influence from other components can also cause variations in the performance and phase profile. For digital system, the delay and fixed point operation make the response a little different from the ideal, floating point design, but may still cause slight variations. Thus, the actual phase response must be measured to accurately compensate for this error.

Frequency Measurement

To compensate the phase in real time, the frequency measurement should be performed in real time. Frequencies f_m and f_r are the frequencies of the input measurement and reference signals, $f_{d,m}$ and $f_{d,r}$ are the Doppler frequency shifts in the measurement and reference signals. Since f_m (f_r) is the sum of split frequency f_s and Doppler frequency shift $f_{d,m}$ ($f_{d,r}$) and the split frequency is nominally known, only one set of frequencies is necessary to measure f_m (f_r) or $f_{d,m}$ ($f_{d,r}$).

There are two potential methods to measure the frequencies. One is the short-time Fourier transform (STFT). This would be suitable to measure f_m and f_r of measurement and reference signals at the beginning of the digital processing in the FPGA. Another method is to directly compute the discrete derivative of the raw phase. This would measure $f_{d,m}$ and $f_{d,r}$ after digital processing getting uncompensated raw phase.

Comparing the two methods, the phase derivative method has better measurement resolution, easier real time implementation and less resource usage. Hence, the derivative method is chosen to monitor the instantaneous Doppler frequency shifts in this design.

Phase Delay Computation

The last main step to compensate the phase delay is solving for the phase delays $\phi_a(f_m)$, $\phi_d(f_{d,m})$, $\phi_a(f_r)$ and $\phi_d(f_{d,r})$. The phase responses $\phi_a()$, $\phi_d()$ and frequencies f_m (f_r) or $f_{d,m}$ ($f_{d,r}$) are already known, thus there are two ways to solve for the phase delays. One is method is a calculation in real time while the another is looking up a table which stores previously calculated phase delays.

Since the LUT quantification error is minimal, controllable, and fewer resources are needed, the LUT was implemented for this work.

We will present results from modeling this effect at different heterodyne frequencies, present real-time measurement techniques for the Doppler frequency, discuss compensation techniques, and present limitations to this approach for measuring and compensating this error.

ACKNOWLEDGMENTS

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A COMPARATIVE PERFORMANCE ANALYSIS OF FDM MACHINES BASED ON A CALIBRATION ARTEFACT

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INTRODUCTION

During the past ten years Additive Manufacturing (AM) technologies have been constantly developing in terms of materials and processes. This allows the use of the AM not only during the preproduction but also for the manufacturing of final components for commercial use [1], [2]. However one of the still existing challenges for AM concerns the quality of the final components. Every manufacturing process has a strict set of requirements that every component has to meet in order to meet production tolerances, yet AM still shows a lack of industrial standards [3]. The advantage of AM to be able to manufacture components of very complex geometries with intricate internal features becomes in this case a drawback. In fact, the control of the quality and the verification of tolerances become difficult task to accomplish with traditional measuring equipment. Some features can be difficult to reach and there are no standards to compare them with.

To overcome this problem, a method to evaluate the performance of AM machine tools based on the printing of an artefact and the subsequent measuring of its features is proposed and shown. This paper shows a validation of the method by means of a laser interferometer. Furthermore, different AM machines are tested using the printed artefact.

METHOD

The performance of the AM machine tools was evaluated using an opto-mechanical hole plate, normally used as a calibration artefact for CMM machines. The plate has dimensions of 120x120x5 mm and it has 25 holes (5x5) with a diameter of 5,5 mm each and a center distance of 20mm in both X and Y directions (see Figure 1) [4].

The plate is printed with an AM machine tool and then measured using a traditional calibration procedure by touch probe CMM [5]. The CMM used in this investigation is a DeMeet 220 with a validated MPE of 4µm. Center coordinates of the

holes are then compared to the nominal values. For positioning evaluation, deviations of printed and measured center coordinates from nominal center coordinates are used.

The AM machine tools and processes involved in this study are listed in Table 1. All machines are based on the FDM process but physical layout and construction differs significantly. The Ultimaker and Witbox are low-end machines, whereas the rest of the machine types can be classified as professional AM machines.

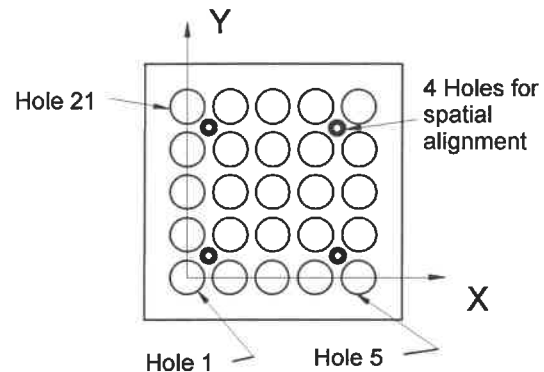


Figure 1 Illustration of the hole plate layout.

Table 1 Machines, processes and materials used for the comparison and evaluation

Machine	Material	Build envelope [mm]
Ultimaker	PLA	210x210x200
Witbox	PLA	297x210x200
Stratasys Dimension 1200	ABS	254x254x305
Stratasys Mojo	ABS	127x127x127

The study is presented in this paper in 3 different steps:

- I. Laser interferometer analysis on a Witbox, (FDM machine). The X and Y axis of the Witbox are analysed using a Renishaw XL80 Laser interferometer. The X and Y axes of the AM machine tool were analyzed based on the principles of ISO 230 [6], but not following the standard in details. A total length of 250 mm for the X-axis and 200 mm for the Y-axis were measured (steps of 10 mm). Subsequently, a hole plate was printed on the same machine and measured using the CMM. Finally the results were compared.
- II. Comparison of different AM machines of the same type. 6 Ultimakers are selected and a hole plate is printed on each of them. Subsequently the plates are measured with the CMM procedure and the data are analyzed.
- III. Comparison of the method applied to different machine types and different processes. 4 different FDM machines (see Table 1) are used in the investigation. A hole plate is printed on each of the AM machines and measured with the CMM procedure in respect of X and Y axis and the hole diameters. Results are then compared.

I. LASER INTERFEROMETER ANALYSIS ON WITBOX

Figure 2 illustrates the performance of the X axis. It shows the mean values of 3 travels on the positive direction, going from 0-250 mm ("X+"), and the mean values of 3 travels on the negative direction, going from 250-0 mm ("X-"). The biggest deviation between the 2 sets of measurements is approximately 200 μm (observed in the very last point of the axis). Standard deviation between the 3 travels in the same direction is approximately 15 μm for the X-axis. The general trend from the laser interferometer is a positive positioning error that could be compensated for. The analysis of the measurements of the printed hole plate in X-direction are also plotted in Figure 2. The points have been placed on the approximate position of the hole plate in the working volume of the printer. The hole plate was printed with axes orientations similar to the machine axes. A first observation is concerned with the fact that the general trend is negative for the hole plate (as opposed to the laser interferometer results). A

second observation is that the order of magnitude of the observed error on the hole plate is comparable with the laser interferometer but with a negative sign.

Subsequently the Y axis was measured following the same principle. Results from the Y-axis are shown in Figure 3. The machine shows a difference between forward and backward movements corresponding to a value of approximately 300 μm . This is caused by the relatively large backlash of the Y-axis. Also a comparison of the results from the laser interferometer and the printed hole plate is shown. Similar trends as for the X-axis can be seen.

When the material is extruded during the movement of the axes, it is deposited onto a fixed build plate. Upon exiting the extrusion nozzle the polymer material cools down and shrinks due to the CTE of the material. If the adhesion of the polymer to the build plate is very good, the effect of the shrinkage is first seen after releasing the printed part from the build plate. The material also expands upon exit of the nozzle due to the pressure difference. Both these phenomena affect the precision and variation of the extruded parts. In particular, the thermal shrinkage of the polymer can explain the different trends in the two graphs. If it is assumed that the material has a CTE of $90 \cdot 10^{-6}$ 1/K then a temperature difference of 40°C (between the material temperature when deposited and room temperature) would allow for a linear shrinkage of approximately 300 μm over a distance of 80 mm. This corresponds to the difference seen in Figure 2. The big backlash of the Y-axis makes a similar analysis of the Y-axis less relevant.

The correlation between the laser interferometer measurements and the hole plate measurements can be established in this way. It is also clear that for the practical use of the AM machines a compensation would be relevant to increase accuracy. This should be based on the hole plate method because the material behavior during printing is included in this approach.

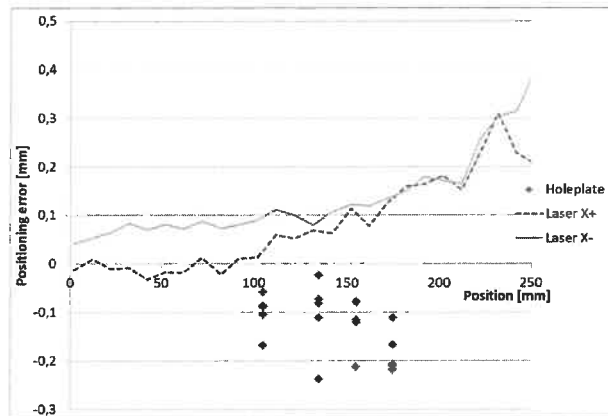


Figure 2 Results from laser interferometer measurements on X-axis.

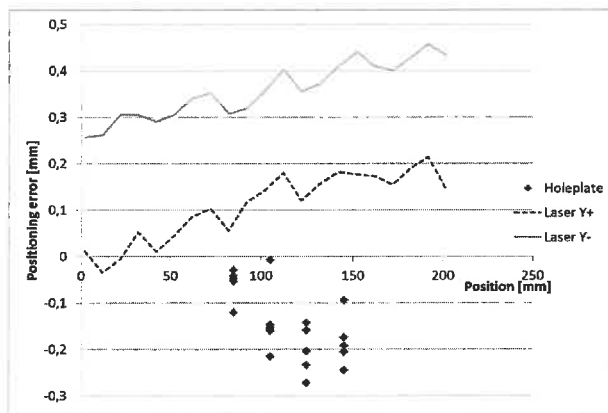


Figure 3 Results from laser interferometer measurements on Y-axis.

II. COMPARISON OF ULTIMAKERS

In order to demonstrate the hole plate as a tool to compare AM machines of the same type, and as a tool to identify type-specific generic errors, a study of six Ultimakers was carried out. A hole-plate was printed on each machine and measured following the proposed method. Figure 4 shows the standard deviation of X and Y axes on each Ultimaker as obtained from the hole plate measurement. In Figure 5 each positioning error is plotted in a two-dimensional grid, from which scale errors as well as squareness errors are visualized.

A general observation from the six machines is that the Y axes show smaller positioning errors than the X axes. This can especially be observed in Figure 4, where data are compared by means of the standard deviation. In Figure 4 it can be observed a much smaller standard deviation on the Y axis meaning that it would be better at reaching the target value. From Figure

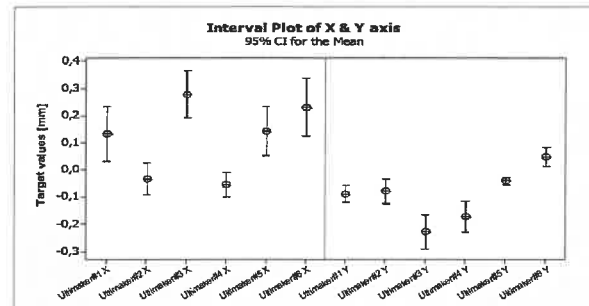


Figure 4 Comparison of the standard deviations of the measurements on the X and Y axis

5 can also be observed that Ultimaker#4 is the one performing the best on X axis while Ultimaker#5 is the one performing best on Y axis. However the machine showing the best overall performance with respect to both X and Y axes is Ultimaker#2. In fact Ultimaker#2 is close to the nominal value and with a small standard deviation on both axes.

If the positioning errors in both directions (X and Y) are compared (see Figure 5) a close similarity on the scale error of the Y axis can be observed between the different machines. Considering that the machines were assembled at the same time by six different people, the trend shown in Figure 5 suggests that this kind of errors is a problem originating from the design and construction of the machine rather than one single machine being assembled in a bad manner.

III. ANALYSIS OF DIFFERENT MACHINES AND PROCESSES

The proposed method is finally used for the analysis and comparison of different FMD printers (see Table 1). Figure 6 shows the results of the analysis of hole plates printed on Ultimaker #1 and Stratasys Dimension 1200 respectively. Clearly the Ultimaker performs worse than the Stratasys machine. Positioning errors as well as squareness errors are significantly larger. This results in a large spread of data with respect to the nominal positions as illustrated at the bottom of Figure 6.

An analysis of the diameters of the holes on the hole plate was performed. The nominal diameter is 5,5 mm. The two Stratasys machines are the closest to target value. So, besides helping rating the machine performance in relation to the others, the calibration artifact also provides information on individual level of each machine with respect to dimensional accuracy in the mm range.

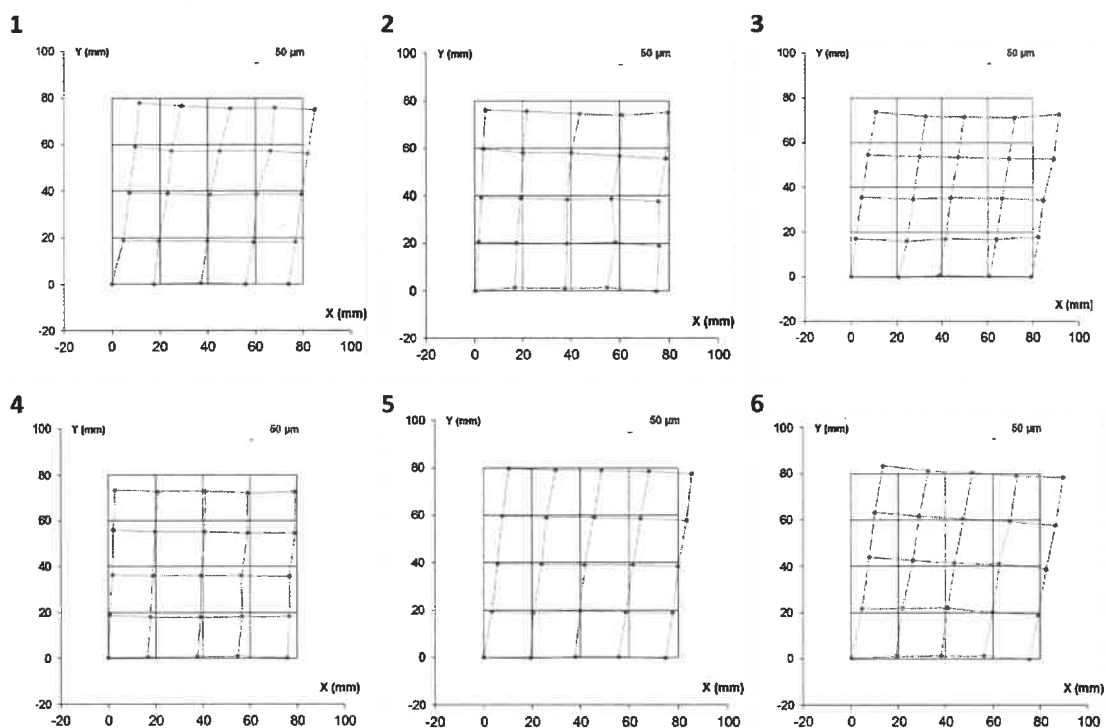


Figure 5 Comparison between the positioning errors of the Ultimakers on X and Y axis

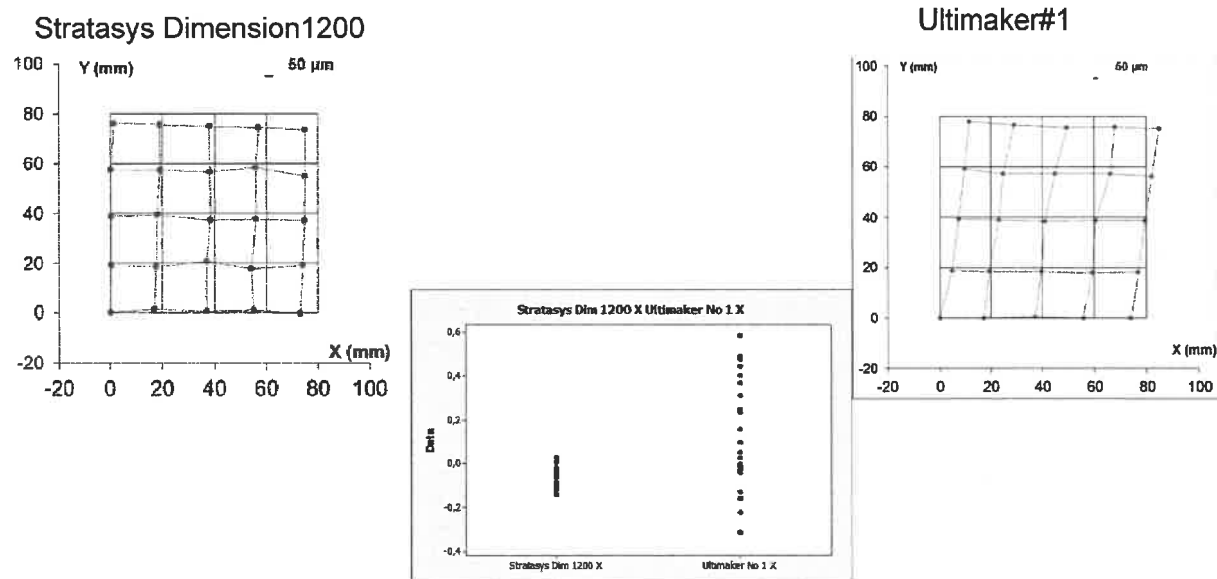


Figure 6 Analysis of two different FDM machines using the hole plate.

Table 2 Diameter comparison of the different printers. Mean and standard deviation [mm]

Machine	Mean	StDev
Stratasys Mojo Diameter	5,535	0,018
Stratasys Dimension 1200 Diameter	5,456	0,014
Ultimaker No 3 Diameter	5,110	0,057
Witbox Diameter	5,445	0,075

CONCLUSION AND OUTLOOK

The proposed approach has proven to be a suitable method for the verification of the performance of AM machine tools. The method was successfully used for the comparison of different machines of the same kind and it allows for an analysis of the errors on the different axis. The method seems to be effective also for the comparison of different types of machines and different processes.

By studying the calibration artifact it is possible to trace the errors in the machine and correct them properly by applying correction factors. Additionally the calibration artifact contains further information, such as the material behavior during the process.

As future work there is the verification of the performance of the machine along the z axis and the application of the correction factor to the Witbox machine for performing further analysis.

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MICRO-LASER ASSISTED SINGLE POINT DIAMOND TURNING ON UNPOLISHED SINGLE CRYSTAL SILICON

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ABSTRACT

In this study Micro Laser Assisted Machining (μ -LAM) is coupled with Single Point Diamond Turning (SPDT) to machine an unpolished single crystal silicon (Si) wafer. Cutting fluids, distilled (DI) water and mineral spirit, were used to decrease tool wear and increase surface quality. Results were very promising for using this process for machining of unpolished Si wafers. It has been shown that combination of laser and cutting fluid not only increases the tool life but also improves the surface quality significantly. Surface finish was improved from 1.2 μ m (as-received) to approx. 80 nm by using this process in optimal condition.

INTRODUCTION

Semiconductors and ceramics such as Si, silicon carbide (SiC), quartz, etc. are very desirable materials in modern industrial applications. These materials have many suitable properties such as being hard, inert, light weight and strong that make them ideal candidates for tribological, semiconductor, MEMS and optoelectronic applications. In most of these applications products require a high quality surface finish and close tolerances to function properly. In spite of all these characteristics, the difficulty during machining and material removal has been a major obstacle that limited the wider application of these materials [1]. Manufacturing these materials without causing any surface and subsurface damage is extremely challenging due to their low fracture toughness. Brittle fracture during processing results in excessively rough surfaces and causes detrimental sub surface damage, which has to be removed in subsequent processing steps. These additional processing steps reduce the overall productivity and

increase the manufacturing cost associated with machining these nominally brittle materials. Machining mirror-like surface finishes contribute significantly to the total cost of a part. In some cases, grinding alone can account for 60-90% of the final product cost [2].

In one of the previous research works [3], it has been demonstrated that the ductile mode machining of semiconductors and ceramics is possible due to the high pressure phase transformation (HPPT) occurring in the material. The ductile response of nominally hard and brittle materials, SPDT/traditional scratch tests were coupled with micro-laser assisted machining (μ -LAM) technique. The μ -LAM system was used to preferentially heat and thermally soften the work piece material in contact with a diamond cutting tool. In μ -LAM the laser and cutting tool are integrated into a single package, i.e. the laser energy is delivered by a fiber laser to and through a diamond cutting tool. This hybrid method can potentially increase the critical depth of cut (DoC), i.e., a larger ductile-to-brittle transition (DBT) depth, in ductile regime machining, resulting in a larger material removal rate.

Those tests in previous works [3,4] were performed on a polished surface of a single crystal Si wafer. Initial study on machining unpolished silicon was very promising, however it was dry machining and tool wear was considerable [4]. The current work focuses more on implementing the μ -LAM technology on a SPDT setup for machining unpolished Si wafers with cutting fluid. The results show that using cutting fluid not only improves the surface finish but also decreases tool wear significantly.

EXPERIMENTAL APPROACH

The Universal Micro-Tribometer (UMT) manufactured by CETR-Bruker Inc. was modified and coupled to the μ -LAM system to perform all of the machining tests. The UMT was primarily developed to perform comprehensive micro-mechanical tests of coatings and materials at the micro scale. Resolution of vertical slider (carriage) of the UMT is $2\mu\text{m}$, therefore lowest programmable depth of cuts was $2\mu\text{m}$.

An IR CW fiber laser, wavelength of 1064nm and max power of 100W with a beam diameter of $10\mu\text{m}$, was used in this investigation. A single point diamond tool with a 1mm nose radius, -45° degree rake angle and 5° degree clearance angle was used for this cutting operation. Figure 1 shows the machining setup used for initial experiments.

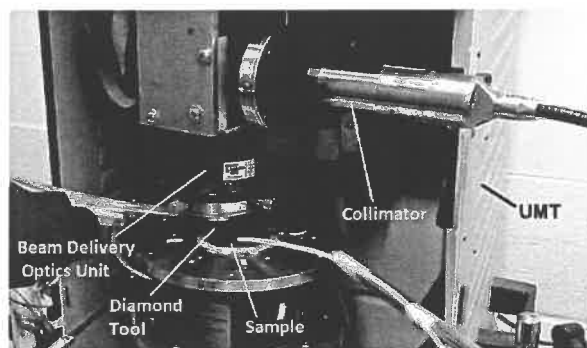


FIGURE 1. SPDT μ -LAM setup for the 2" Si wafer

DI water and odorless mineral spirit, as two common types of cutting fluid, were used for machining Si wafers in this study. Since laser was using, to avoid any fire during process a nonflammable mineral spirit was used.

In this research effects of different parameters such as using cutting fluid, laser power and cross feed rate, as table 1, on process outputs like surface roughness (R_a) have been investigated. These parameters have been chosen based on previous results. To understand the effect of using laser and cutting fluid during process, Si samples machined with/without laser with those two cutting fluids. Note that laser power output after diamond tool was about 40 % of adjusted power due to scattering, reflections, absorption and etc. It means for adjusted laser power of 20 W , actual output was about 8 W . All tests were performed

at constant spindle rotational speed of 100 RPM . All other conditions were kept constant to have same conditions for all tests.

Initial tests - on polished surface of a Si wafer - showed that the micro-laser assisted SPDT tests were successful in demonstrating the benefits of the thermal softening effect from preferential laser heating [3-5].

Table 1. Parameters used for machining

Laser Power (W)	Cutting fluid	Cross feed ($\mu\text{m}/\text{rev}$)
0	DI water	30
		10
20	Mineral spirit	2

The focus of this study is on machining an unpolished single crystal Si wafer which is very brittle and hard to machine by conventional methods. Figure 2 shows microscopic image of unpolished surface of a Si wafer and also its three-dimensional image captured by using a white light interferometer. As it is clear, surface is full of peaks and valleys. The aim of this work is to reduce Si wafer roughness as much as possible with least number of machining passes with the lowest time of machining.

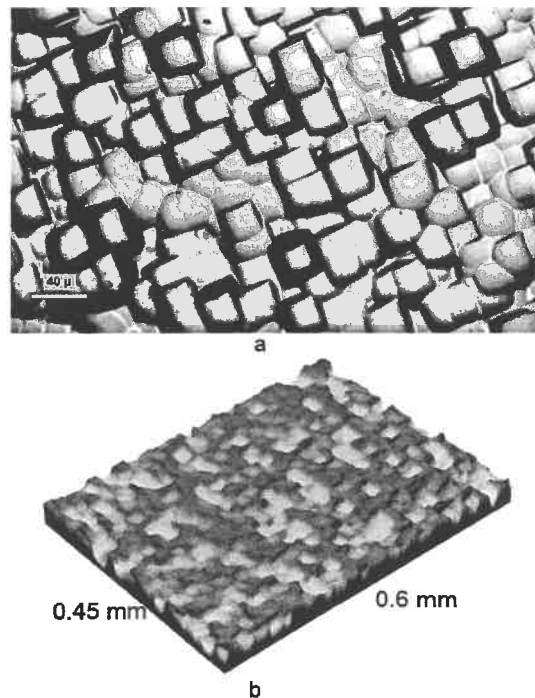


FIGURE 2. a) Microscopic and b) 3D image of unpolished region of a Si sample

RESULTS AND DISCUSSION

All the cuts performed in this experiment were in ductile regime. Feed marks on machined surfaces are a sign of ductile mode machining because a brittle material like Si only in metallic phase can flow. The surface roughness was measured by using the white light interferometer and the tool wear was investigated with an optical microscope.

Figure 3 shows microscopic images of the machined surface of the Si without using laser with 30, 10 and 2 $\mu\text{m}/\text{rev}$ cross feed rates. Mineral spirit used as cutting fluid for these tests. Dark areas in this figure represent either remained valleys of surface or brittle mode machining that should be removed in final machining process. White areas in this figure represent ductile mode machined areas. Figures 4 and 5 show the machined regions with laser using mineral spirit and DI water as cutting fluids respectively.

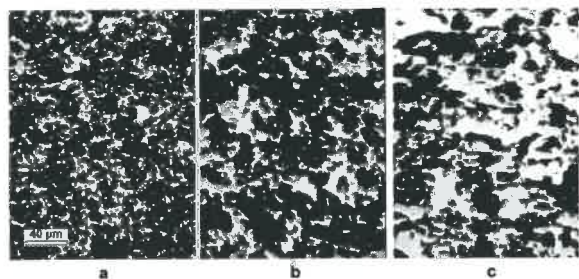


FIGURE 3. Microscopic image of machined surface of a Si wafer with no laser and mineral spirit as cutting fluid with a)30 b)10 c)2 $\mu\text{m}/\text{rev}$ cross feed rate.

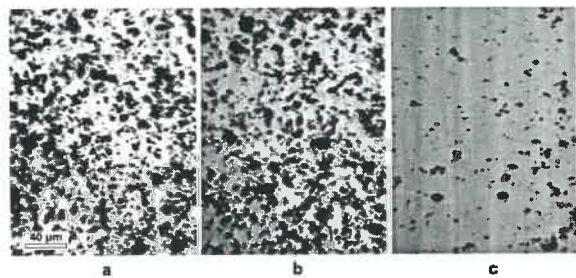


FIGURE 4. Microscopic image of machined surface of a Si wafer with laser and mineral spirit as cutting fluid with a)30 b)10 c)2 $\mu\text{m}/\text{rev}$ cross feed rate.

As it is clear from images 4 and 5, surface qualities are much better than when no laser is used. Regions machined with 30 and 10 $\mu\text{m}/\text{rev}$ cross feed still have dark areas; however they

are much smaller than no laser condition. Surfaces machined with 2 $\mu\text{m}/\text{rev}$ cross feed are very smoother and for example for the sample machined with DI water is almost in pure ductile mode (Figure 5c).

The results from the analyses presented in figure 6 suggest that the surface roughness's for all regions with laser and cutting fluid are better than that of no laser power (room temperature). It was found before that higher depth of cuts can be achieved by using laser which directly translates to increased material removal rates and higher productivity [5,6]. In addition, the cutting forces for the machined regions with laser are lower suggesting extended tool life for large scale production.

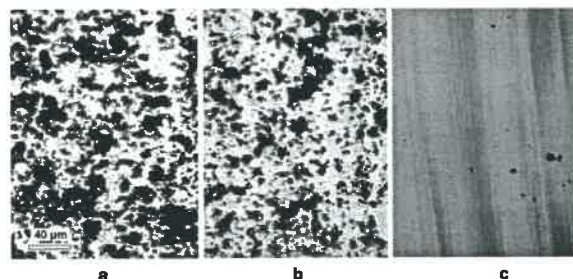


FIGURE 5. Microscopic image of machined surface of a Si wafer with laser and DI water as cutting fluid with a)30 b)10 c)2 $\mu\text{m}/\text{rev}$ cross feed rate.

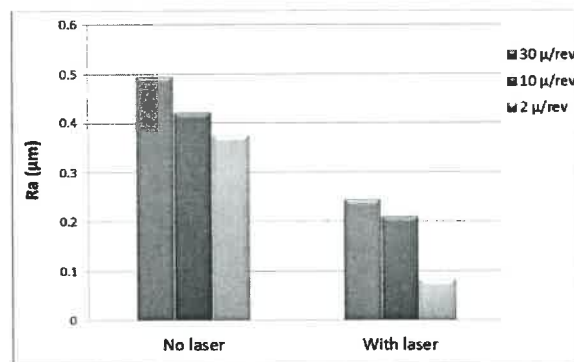


FIGURE 6. Surface roughness (Ra) corresponding to sample machined with DI water with and without laser

The surface roughnesses of samples machined with different cutting fluids are presented in a graph in figure 7. Even though in rough pass (30 $\mu\text{m}/\text{rev}$) Ra of Si sample machined with mineral spirit is lower than that of with DI water, for low feed rate (2 $\mu\text{m}/\text{rev}$) it is vice versa. Ra for 10 $\mu\text{m}/\text{rev}$ for both conditions is almost same.

In optimal condition the surface roughness can be improved from 1.2 μm (as-received) down to approx. 83 nm. Achieving such a surface finish from an unpolished surface is a very promising result for this process. Also tool wear during the tests using laser power was less than when no laser has been used. Results show that using cutting fluid, in this study DI water and mineral spirit, not only improves the surface finish but also decreases the tool wear significantly.

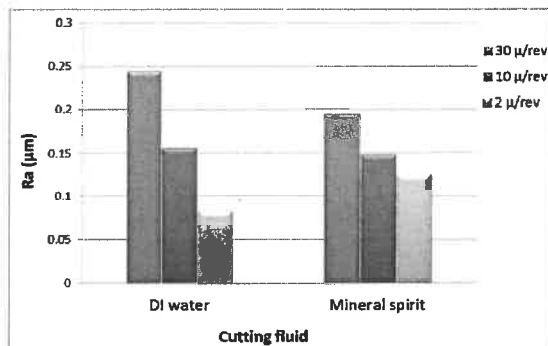


FIGURE 7. Surface roughness (R_a) corresponding to different cutting fluid and cross feed rate

Figure 8 shows a 3D image of Si sample machined with no laser with mineral spirit as cutting fluid. This is the same surface as figure 3c, which shows those black areas in figure 3c were actually pits. As mentioned before these pits can be either the remained valleys of original surface or because of ductile to brittle mode transition.

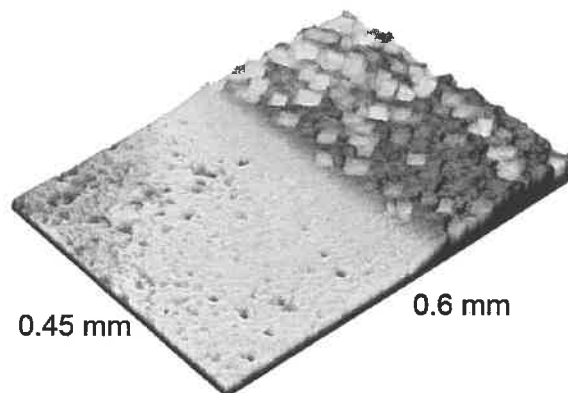


FIGURE 8. 3D image of unmachined region (right) and machined region with 2 $\mu\text{m}/\text{rev}$ cross feed rate with no laser (left).

In figure 9 an unmachined area is compared to a surface machined with 30 $\mu\text{m}/\text{rev}$ cross feed rate with laser. Even though cross feed rate for this region is much higher than the region machined

with 2 $\mu\text{m}/\text{rev}$ in figure 8, surface is much better due to using laser. Comparing figure 8 and 9 shows how much laser can improve the surface quality.

In figure 10 a surface machined with same condition as surface in figure 8 (2 $\mu\text{m}/\text{rev}$ cross feed rate) but with laser is compared to an unmachined area. It is same surface as figure 5c. Almost no sign of brittle mode or transition can be seen in this image and it is a pure ductile mode machining.

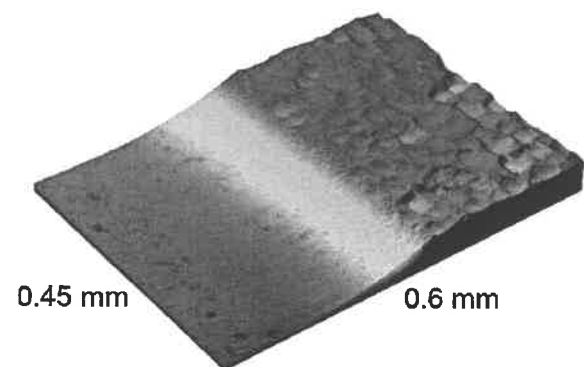


FIGURE 9. 3D image of unmachined region (right) and machined region with 30 $\mu\text{m}/\text{rev}$ cross feed rate with laser (left).

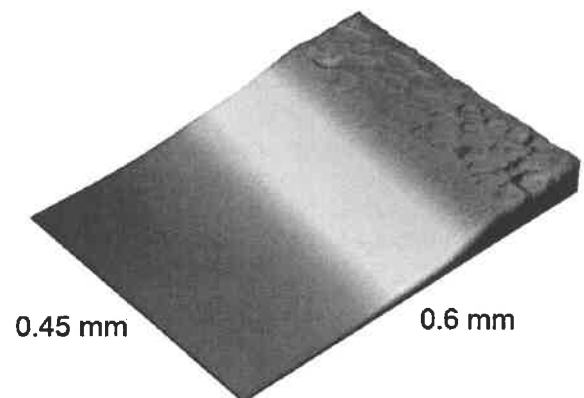


FIGURE 10. 3D image of unmachined region (right) and machined region with 2 $\mu\text{m}/\text{rev}$ cross feed rate with laser (left).

Tool wear after machining all Si samples (figure 11) and track length of about 7 km was very minimal (compared to previous work [6]). It seems combination of laser and cutting fluid improved the tool life significantly.

SUMMARY

Results of this study show that the micro-laser assisted SPDT tests were successful in

improving surface finish of unpolished Si wafers and less tool wear. Best surface roughness, $R_a = 83$ nm, obtained with laser and DI water as cutting fluid, while R_a for sample machined with no laser with cutting fluid was 376 nm. The effect of a combination of laser and cutting fluid on surface roughness is obvious and promising.

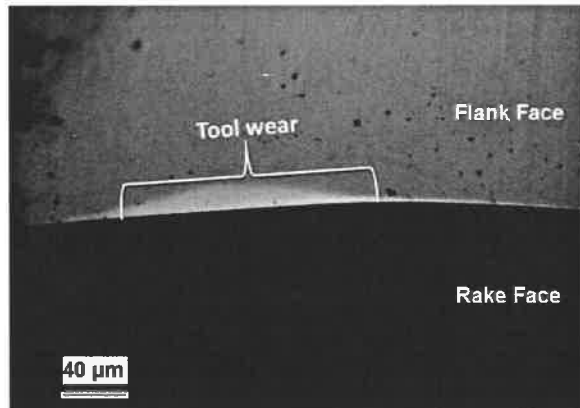


FIGURE 11. Microscopic image of tool after 7 km machining track length.

ACKNOWLEDGEMENT

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EFFICIENT MICRO-CONCENTRATOR FOR MICROSYSTEMS-ENABLED PHOTOVOLTAICS

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MOTIVATION

In response to the DOE SunShot initiative to achieve utility scale solar power generation at \$1 per watt peak, Sandia National Laboratories is investigating a microsystems-enabled photovoltaics (MEPV) approach that combines the high conversion efficiencies of concentrated photovoltaics (CPV) with the form factor and low system costs of flat panel PV. MEPV combines solar cells having lateral dimensions smaller than 1 mm with micro-optic concentrators to reduce the use of expensive semiconductor materials and to increase solar conversion efficiency [1]. While existing semiconductor infrastructure is being leveraged for MEPV cell research [2], a parallel effort is underway to develop design and fabrication techniques for MEPV micro-optics.

DESIGN

Previous work has described the progress of prototype systems [3,4] as micro-optics have been developed with a 100X magnification and a $\pm 2.5^\circ$ field of view for 250 μm diameter multi-junction "stacked" MEPV cells. Current work has developed a third generation prototype with a 200X magnification and a $\pm 1.7^\circ$ field of view in the visible and NIR spectrum. The optical design, Figure 1, was simplified from the first two prototypes to involve a single PDMS lens array which is cast directly onto a silicon substrate containing the MEPV cell array. The wavelength band of the optics is roughly 400 to 1600 nm as different regions of the solar spectrum are collected by different layers in the stacked cells. Optical system complexity has been reduced with each successive prototype in an effort to reduce material and fabrication costs. In this design, a single plano-convex lens is used for concentration which incorporates an

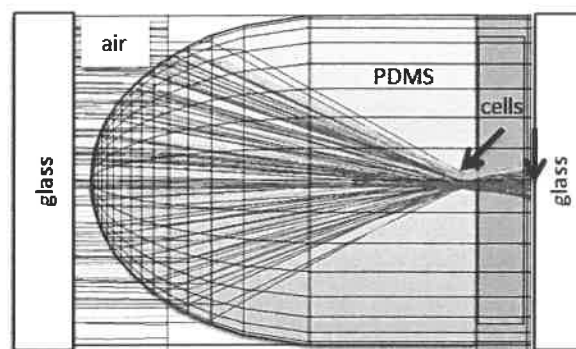


FIGURE 1. MEPV optical design.

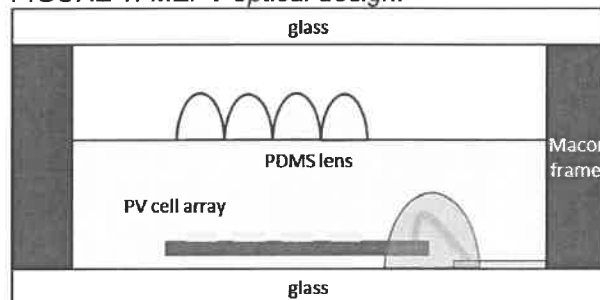


FIGURE 2. MEPV module cross-section.

8th order asphere with a conic term to minimize cell losses due to local hotspots on the cell surface. Its aperture diameter is 3.85 mm as the lenses are arranged in a hexagonal closed pack pattern with 120 lens elements for a footprint size of roughly 50 mm square. Alignment requirements have been simplified from prior designs to involve only the alignment of the cast lenses to the MEPV cells. The silicon substrate and PV cell array is mounted onto a 3 mm thick, glass backplane to provide strength and an environmental barrier, Figure 2. The module is enclosed by a Macor frame that more closely CTE matches the glass than metal and provides the standoff for the front protective glass.

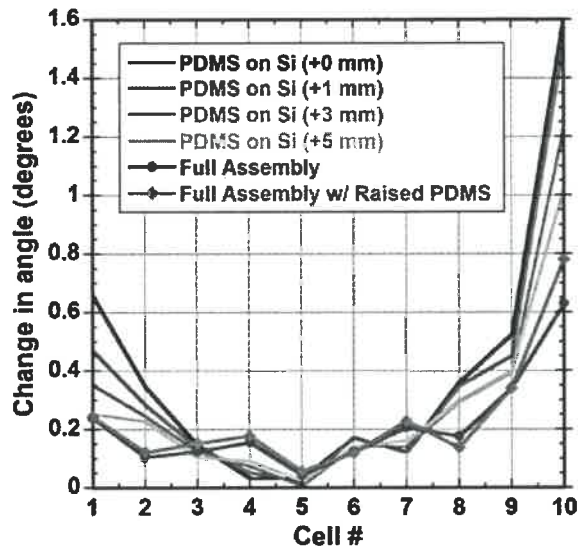


FIGURE 3. Lens tilt across an array for a 60° C temperature change.

Thermo-mechanical modeling was performed during the design phase to insure module performance across its anticipated operating temperature range of -40° to 80° C. The distortion of the front glass due to air volume changes with temperature was analyzed to prevent damage. Volume changes were determined based on a simple ideal gas law relationship, $P \cdot V = n \cdot R \cdot T$, and used to predict glass deformation. Front glass thickness had the greatest impact on deformation and resulted in a design thickness of 3 mm. Shear and normal stresses in the PDMS lenses from curing at 80° C were also analyzed to insure that delamination would not occur. Finally, simulations of lens distortion for temperatures from 80° to 20° C revealed that adhesion of PDMS material to the module sidewalls is critical to reducing lens array warping. Figure 3 plots the tilt angle of lenses in one direction across the array where its center is fixed. Minimum lens distortion occurs when the PDMS bonds directly to the module frame but does not extend above the lens surfaces.

ASSEMBLY AND FABRICATION

The first two prototype MEPV modules contain significant complexity and introduce a host of challenges to achieving low cost, high volume production. Recent work has leveraged prior experience to further reduce module complexity. Precision alignments are only required during the first two module assembly steps where the silicon substrate is attached to the glass backplane and then the PDMS lenses are cast

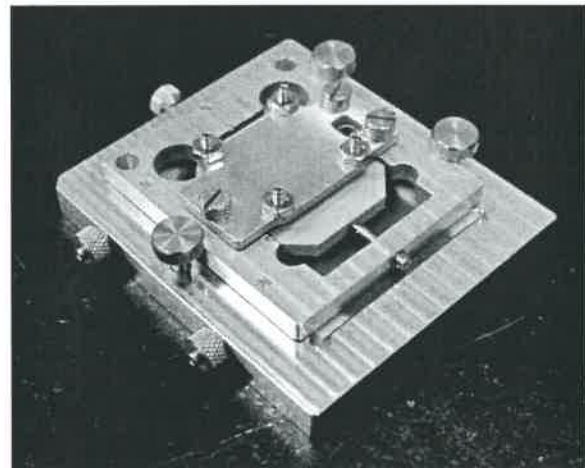


FIGURE 4. Fixture for assembling the silicon substrate to the rear glass backplane.

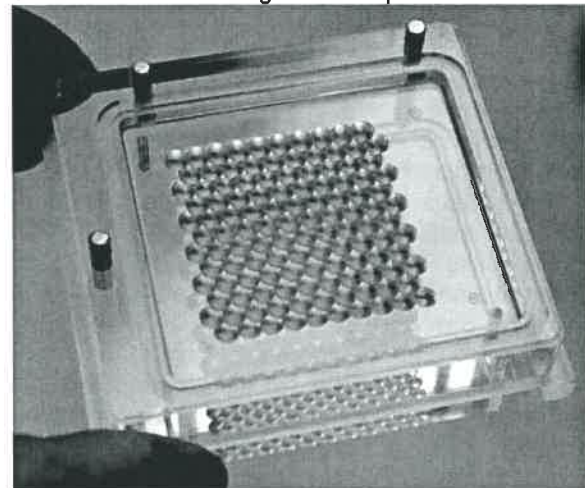


FIGURE 5. Acrylic test mold.

onto the Si-glass subassembly. Figure 4 shows the mechanical fixture used to align and mount the silicon substrate to the glass backplane. All assembly is referenced based on ground edges of the glass as the fixture aligns the Si to the glass using dowel pins. The silicon is adhered to the glass using PDMS. Mounting and bonding of Si substrates without MEPV cells has been demonstrated with an accuracy of 5 μm . While the mechanical fixturing utilized during product development are not scalable for mass production, the alignment principles utilized and general process flow could be implemented in production equipment. Casting of the PDMS lens array involves a mold that is also aligned to the ground edges of the glass using dowel pins. The final aluminum mold will be produced by rough micro-milling and finish diamond precision milling and is still in process. An acrylic mold, Figure 5, however, was micro-milled and has been used for process development. Casting of

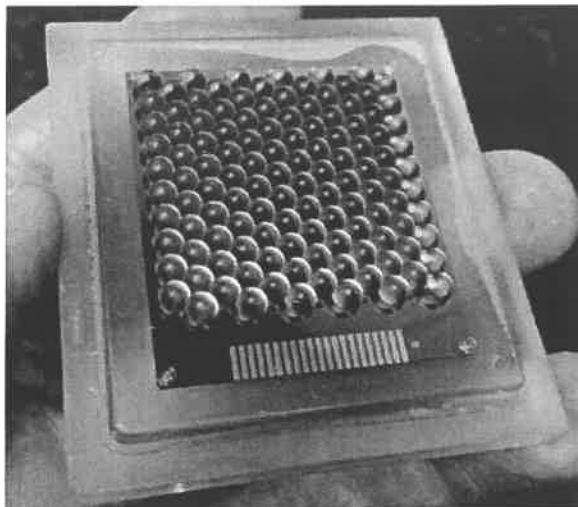


FIGURE 6. Lens array cast onto a Si-glass subassembly.

a 120 element, PDMS lens array has also been demonstrated onto a Si-glass assembly using the acrylic mold, Figure 6, with lens to cell alignment accuracies on the order of 10 μm . Wire bonding the cell array to the glass backplane metal traces, mounting the Macor frame, back-filling the frame to lens gap with PDMS and mounting the front glass to the frame do not require precision assembly, are relatively straightforward, and have been demonstrated.

FUTURE WORK

Process development for a third generation MEPV system continues with functional module assemblies expected in the coming weeks. Final process and module performance will be available for presentation at the annual meeting.

ACKNOWLEDGEMENTS

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DESIGN AND FABRICATION OF AN OPTICAL SYSTEM FOR A BALLOON-BORNE SPACE TELESCOPE

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INTRODUCTION

The NASA Balloon Experimental Twin Telescope for Infrared Interferometer (BETTII) concept, shown in FIGURE 1, consists of two identical telescopes mounted onto a space frame and carried to an altitude of 130,000 ft. by a balloon. Each telescope has four optical elements – a primary mirror, a turning flat, a secondary mirror, and a tertiary mirror.

The balloon-based mission brings with it many unique challenges for designing, fabricating and assembling the optical system. During the ascent into the high atmosphere, the telescope will experience temperature changes of 250 degrees F. To minimize the effect of these changes, the entire telescope is Aluminum 6061-T6 so that the optics will all expand and contract the same amount during this thermal cycling. In addition, the Cryostat with the optical sensors is hanging from the telescope (see Figure 2) to assist beam alignment and the telescope/cryostat is supported by the Metering Truss. The telescope is tilted 13.3 degrees with respect to the horizontal to allow full optical access to the primary mirror from the siderostat.

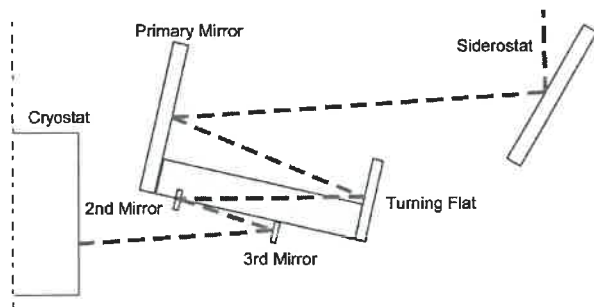


FIGURE 2. Layout of the telescope optical system

TELESCOPE DESIGN

The design and fabrication of the two identical telescope assemblies shown in Figure 2 are the subjects of this paper. The primary mirror is a

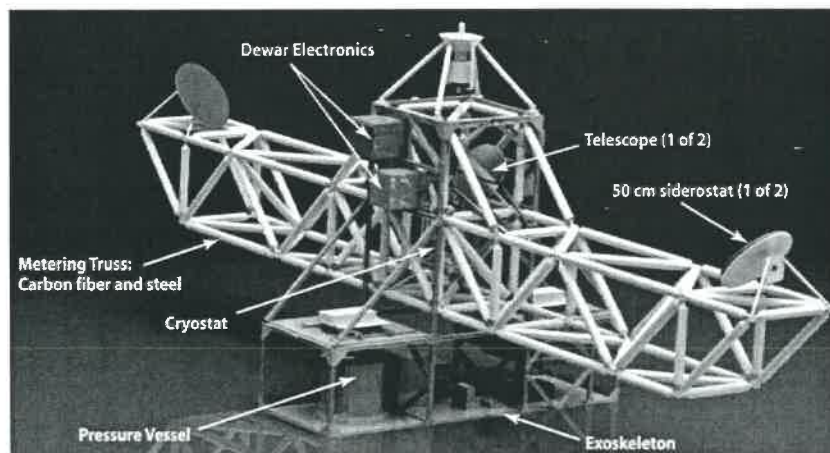


FIGURE 1: BETTII Infrared Binocular Space Telescope [1]

segment of a 2.75 m parabola that is 651 mm off-axis with an aperture of 522 mm, as shown in FIGURE 3. In its lightweight form, it has a mass of 5 kg. The mirror face is diamond machined on center using a fast tool servo to produce the non-rotationally symmetric tool path that will create the off-axis shape. The flat, 220 mm diameter turning mirror in Figure 2 is supported from the primary through a thin walled "trough" mounting structure. The two other mirrors in each telescope are complex, non-rotationally symmetric designs but are small (< 40 mm) and can be seen sticking through a slot at the bottom of the trough in Figure 2.

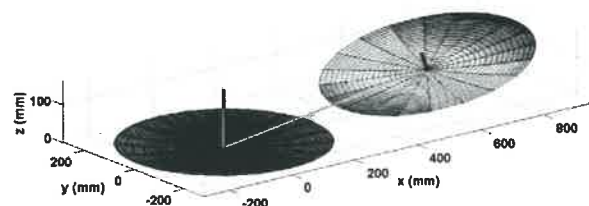


FIGURE 3: Primary mirror optic at parent location (right) and centered/tilted for machining

The design challenge was to keep the overhanging 2 Kg load caused by the turning flat from distorting the primary while robustly supporting the flat mirror which is 800 mm from the primary. This is accomplished by the combination of mounting rings that are bolted to each end of the trough and kinematic support of the two mirrors via Kelvin

couplings. Both of the telescope assemblies are mounted to the steel/carbon fiber space frame by struts attached to the mounting ring shown in Figure 2. The struts have flexures that allow the aluminum mirrors and trough to contract at a different rate than the space frame without imparting significant moments into the optic structure.

Primary Mirror

Figure 4 shows the design of the primary mirror with light-weighting ribs on the back and toroidal features on the front that form half of the non-influencing kinematic mount. Raised bosses on the back of the mirror will be used along with spherical washers to hold the part during diamond turning. The mirror has light-weighting features on the back consisting of thicker ribs with a wide spacing to combat gravity sag and thin ribs spaces 30 mm apart to reduce the amplitude of print-through during diamond turning.

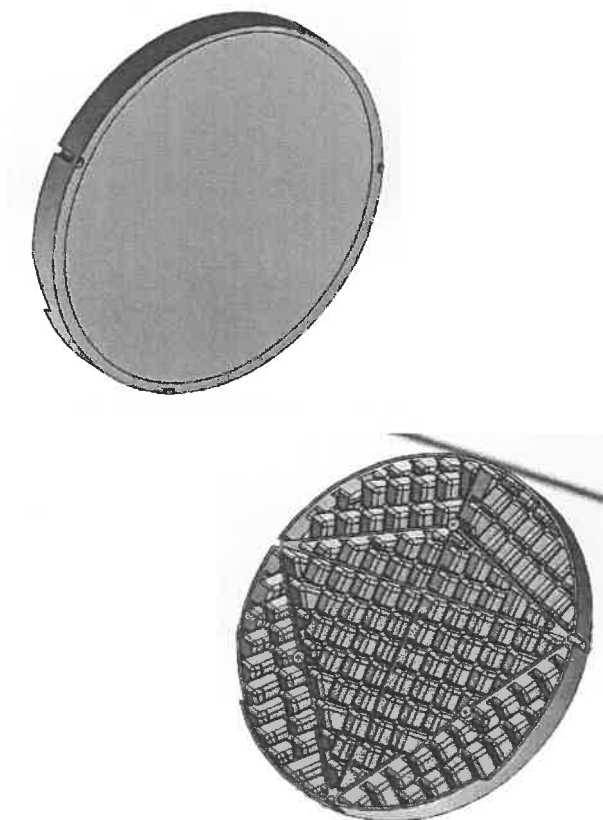


FIGURE 4: Primary mirror front (top) with kinematic coupling and back (bottom) with light-weighting ribs

Analysis showed that gravity will cause the primary to sag by 426 nm PP when tilted 13.3 degrees, as shown in FIGURE 5. Since this telescope is

designed to study long wave-length ($> 1.5 \mu\text{m}$) light, form errors on the order of 400 nm are acceptable.

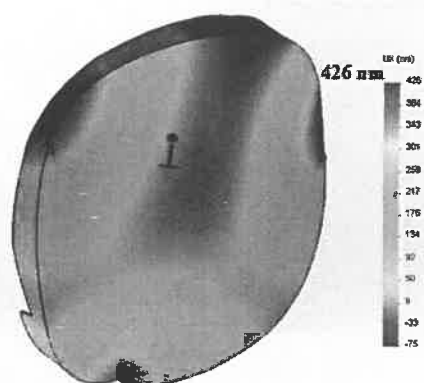


FIGURE 5: Deflection of primary mirror due to gravity with PV of 426 nm.

Machining Primary Mirror

The primary mirror is machined on a Nanoform 600 DTM coupled with the FLORA II fast tool servo built at the PEC. The off-axis parabolic section is machined on axis by translating the shape to the center of rotation, tilting it and finding the best fit rotationally symmetric surface with a sag of 13 mm for the DTM to follow as shown in Figure 3. The 1 mm correction for the non-rotationally symmetric surface, shown in FIGURE 6, is simultaneously added by the FTS. The result is an optical surface with form error of < 200 nm RMS and surface finish less than 30 nm RMS.

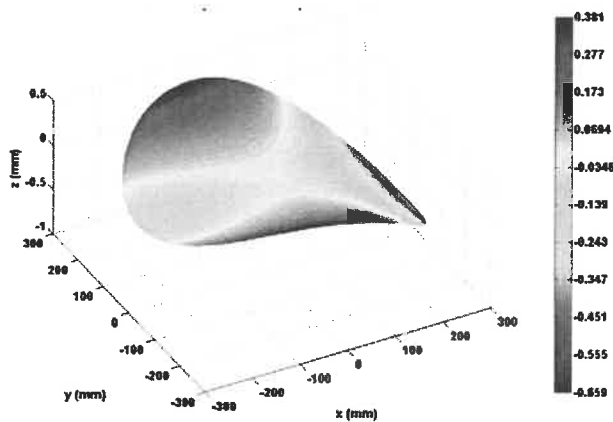


FIGURE 6: Non-axially-symmetric component of OAP after best fit sphere of radius 2.92 m is removed. FTS motion is approximately 1 mm.

Because the surface of the mirror is 2.5 mm thick, print-through due to tool thrust force is a concern. Analysis was done by applying 1 N force at many locations on the front of the mirror and calculating

the deflections. The sum profile – approximated by placing an equivalent pressure load on the front face – is shown in Figure 7.

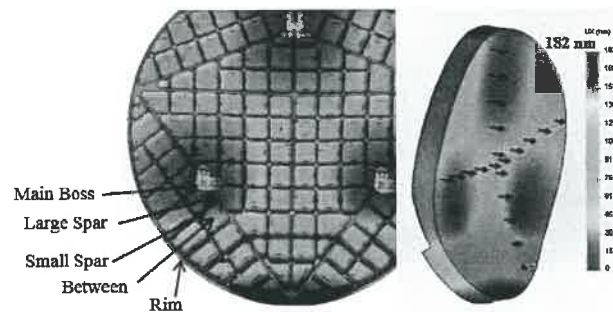


FIGURE 7: Deflection due to tool thrust force at different locations on the primary mirror with PV=182 nm

TABLE 1: Predicted deflection due to 1 N tool thrust force at different locations

Location	Deflection (nm)	Resulting Error (nm)
Main Mounting Boss	32	0
Large Support Spar (near boss)	39	7
Outer Rim	173	141
Small Support Spar	85	53
Between Spars	118	86

TABLE 1 shows that FE model deflection at a number of different points on the face of the primary as a result of a 1 N thrust force at that point caused by the diamond turning operation. The second column assumes that if the load is applied everywhere for the tool pass, each of these deflections will be achieved and the shape will have print-through features less than 141 nm PP.

Turning Flat

The Turning Flat poses similar challenges to the primary but with a smaller diameter. The flat mirror uses the same toroid-shaped feature for its kinematic mount as the primary. The back of the flat mirror is light-weighted, but because it is half the size of the primary, the simple pattern shown in FIGURE 7 is sufficient to combat gravity sag as well as print-through. Also, the features on the back side of the flat mirror can be cut out with a square end mill, eliminating the need for weight-adding fillets at the bottom of the pockets. The outside of the Turning Flat is 254 mm in diameter and 12.7 mm deep. The face is 2 mm thick, the outer rim is 4 mm wide, and the rest of the ribs on the back are

2 mm wide. As with the primary, raised bosses are used for mounting during diamond turning.

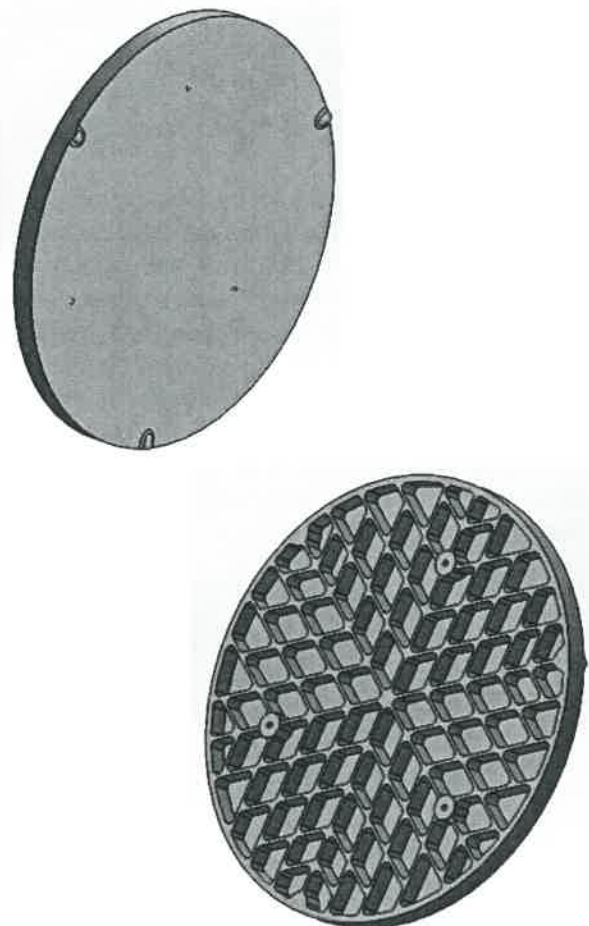


FIGURE 8: Front (top) and back (bottom) of the Turning Flat. Note the kinematic coupling features.

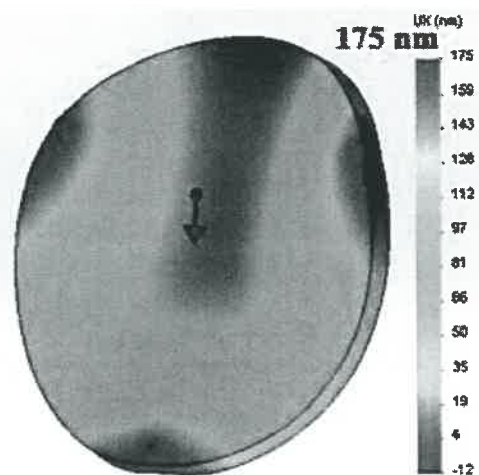


FIGURE 9: Deflection of Turning Flat due to gravity sag

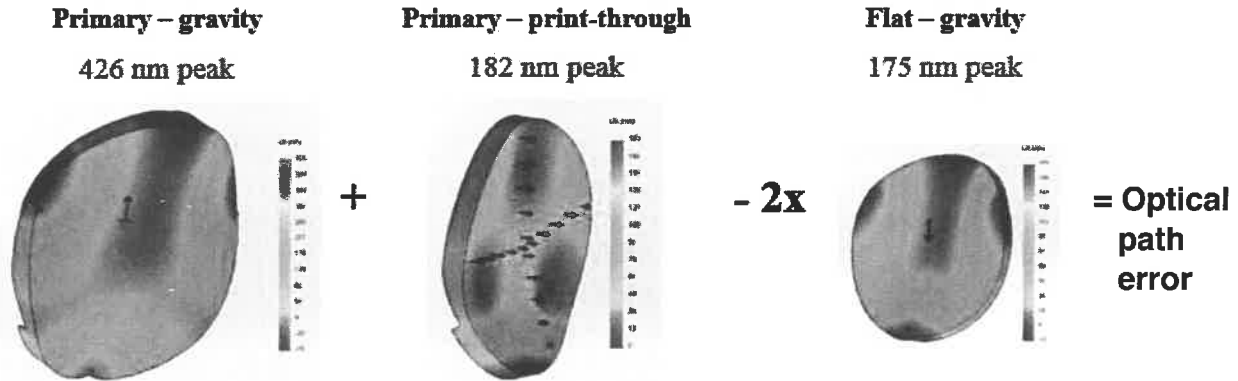


FIGURE 10: Effects of different form errors on optical path

Combined Effect of Form Errors

Since the flat mirror is nearly parallel to the primary and supported with 3 features around its rim, the shape of its deflection is qualitatively similar to that of the primary, as shown in FIGURE 8, with about half (175 nm) the peak amplitude. Both the primary and the secondary sag in the same spatial direction, so that their effects on the optical path will cancel. The predicted form errors are shown side-by-side in Figure 10. Note that deformations of the turning flat have a two-fold effect on optical path length.

MOUNTING STRUCTURE

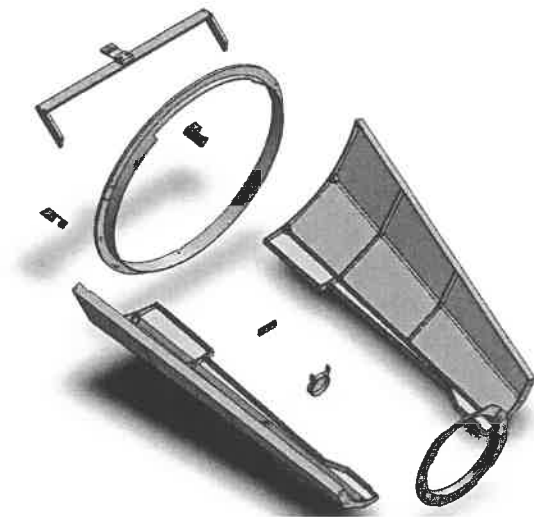


FIGURE 11: Mounting structure with thin-walled trough and coupling rings

The components in the mounting trough-ring structure are decomposed in FIGURE 11. Each part is machined from a single piece of aluminum 6061-T6 and heat-treated to relieve internal stresses. The two sides of the trough are joined in the middle first to produce a rigid one-piece base.

The rings are then bolted to the ends to create the optical bench. The mirrors are mounted to this reference frame using the non-influencing kinematic couplings and spring tabs. The alignment of the primary mirror and the turning flat can be measured and adjusted by shimming the four bolted connections between each ring and that end of the trough.

The mirrors are attached to the rings using a kinematic Kelvin coupling that enables repeatable positioning and decouples deformations in the mounting structure from the optical surface. The Kelvin coupling consists of a toroidal feature that is machined on the front of each mirror and a triangular groove cut into the mounting rings, as shown in Figure 12. The mirrors are then clamped through the center of the toroid features using a constant-pressure clamp.

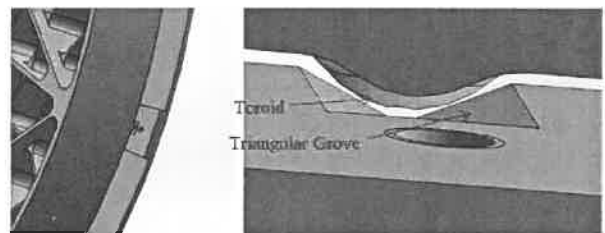


FIGURE 12: Kinematic Kelvin coupling

ASSEMBLY AND ALIGNMENT

Once the primary and turning mirrors are aligned, the telescope is mounted to the x-frame that attaches it to the Metering Truss. The Cryostat is hung from the primary mirror mount and the remaining mirrors are adjusted to get the collimated beam into the cryostat for analysis.

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ANALYSIS OF NOZZLE DESIGN USED FOR THE CREATION OF ADVANCED ENERGY BEAM

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INTRODUCTION

A variety of scientific and industrial projects, such as segmented ground based telescopes, compact space based observers, short wavelength microlithography and high power laser systems, demand metre scale ultra-precise surfaces [1]. Cranfield University and Loxham Precision have been engaged in developing effective fabrication of medium to large optical surfaces for the aforementioned applications. A process chain of three sequential machining steps has been proposed (Figure 1). These steps are ultra-precision grinding, robot based polishing and plasma figuring. The fabrication target is to reach a 20 hours cycle time for each stage of surface generation for 1.5m size optics: equating to 1ft² per hour [2-3].

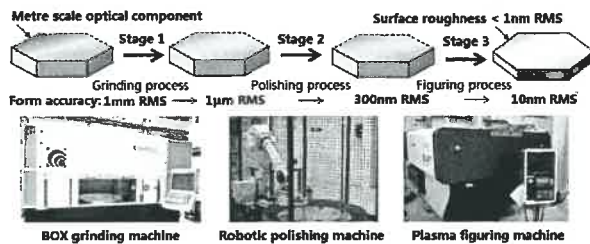


FIGURE 1. Proposed rapid production line for large scale mirror segment fabrication [2].

BACKGROUND

The final processing step of this fabrication chain is a state-of-the-art plasma figuring process. This figuring process uses an Inductively Coupled Plasma (ICP) torch whose motion is managed by the dwell time algorithm that provides a controllable chemical reaction for etching local regions of a surface. The use of an ICP torch for the study of surface aberrations was carried at Lawrence Livermore National Laboratory (LLNL) in the late 1990s by Dr Jeffrey Carr et al [4-7]. From this work a series of processing systems were created. The measurement of the plasma temperature has been carried out by Dr William O'Brien [8]. The

largest and most sophisticated plasma figuring machine was created in 2008 through a collaboration between RAPT Industries of the US and Cranfield University in England. The so-called Helios 1200 is a unique 1.2 metre scale plasma surface figuring facility housed within the Cranfield University Precision Engineering Institute [9]. In 2012, Castelli et al [10] demonstrated fast figure correction of near meter scale optical surfaces to 30nm RMS form accuracy using this plasma technology.

This large scale plasma facility uses an ICP torch which is operated at atmospheric pressure. The plasma torch is able to create a highly collimated plasma beam, in which fluorine atoms are introduced. These fluorine atoms react with silicon-based substrates. Typically, material removal rate of ~1.5 mm³/min is achieved. The material removal footprint of the ICP torch is characterised by a Gaussian cross section with "soft edges". An important feature of the plasma torch is its dedicated nozzle. This nozzle is mounted onto the end of the ICP torch. The nozzle design determines the size, shape and plasma velocity of the energy beam as it impinges the processed surfaces.

The processing of metre scale optical surfaces achieved in 2012 was successful both in terms of processing duration and form accuracy. Plasma figuring of a 440mm sized substrate was performed in less than 2.5 hours achieving 30nm RMS form accuracy from an initial 2.5 micrometre PV value. However, mid spatial frequency (MSF) structure was evident [11-12]. This surface structure was assessed and linked to the raster-scanning parameters and ICP torch nozzle design. Figure 2 highlights this surface structure showing the main spatial frequency and its harmonics.

The identified and undesired residual surface features need to be removed. To accomplish

this improvement, the use of smaller scale energy beams has been suggested [12]. The energy beam footprint achieved in the previous research is shown in Figure 3. It can be seen that Full Width at Half Maximum (FWHM) is 12mm.

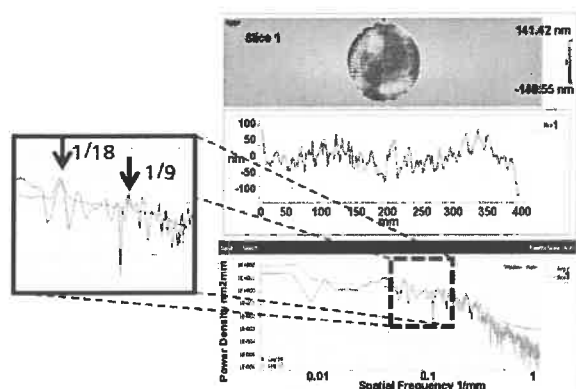


FIGURE 2. Surface structure showing MSF features [11-12]. (In 2012, Jourdain and Castelli scrutinised the MSF on a 400mm diameter surface processed by plasma figuring process)

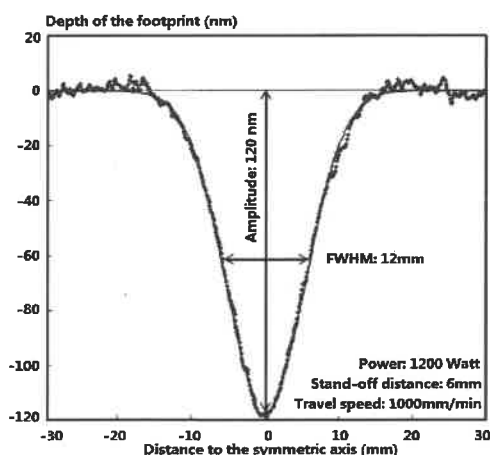


FIGURE 3. Footprint of the trench (cross section)

Numerical simulation of the plasma torch nozzle is considered essential to investigate and understand the physical and chemical processes taking place. Previously, Professor Boulos and Professor Proulx have created computational fluid dynamics (CFD) models for subsonic and supersonic ICP torch systems [13-16]. The fluid description of the plasma was based on the Navier-Stokes conservation equations for mass, momentum and energy. Many properties of the plasma and its behaviours in the ICP torch have been investigated through those models.

The purpose of the work presented in this paper is to advance the plasma figuring of optical surfaces through the development of optimised ICP torches and associated nozzles. This research aims at providing highly collimated energy beams characterized by a material removal footprint of between 1mm and 5mm FWHM.

This paper describes initial work towards the creation of a CFD model of the plasma process with initial attempts to correlate CFD results with plasma beam removal footprint data. This CFD model is developed to help understand some important design rules for new nozzles.

NUMERICAL SIMULATION OF PLASMA NOZZLE DESIGNS

This paper introduces initial CFD modelling of the plasma torch nozzle designs. As a first study the CFD model evaluates the aerodynamics of the gas flow. A number of assumptions are made regarding the fluid. The fluid is considered to be high temperature argon gas, it is also assumed axisymmetric, uniform, steady and laminar. Consequently, a simple 2D cross-sectional model of De-Laval nozzle design has been created. This model is based in the software package FLUENT (Figure 4).

The CFD model boundary conditions include: flow input velocity, gas type, gas temperature, and pressure distribution. The initial entry temperature is set at 6000 Kelvin as supported by the PhD work of Dr O'Brien.

Figure 4 shows a 3D drawing of the ICP torch and a cross-section of the current nozzle as modelled in the FLUENT software. Argon is fed into the nozzle through the upper aperture and flows downward in an axis-symmetric manner. Two areas are of interest in this work: the De-Laval throat and the near surface substrate region where the chemical reaction takes place. This second area is the focus of this paper.

To ease explanation a line of study is defined at 10 μm from the substrates surface. This line is entitled the "Pathway of Investigation", see Figure 4.

As shown in Figure 5 the "Pathway of Investigation" is characterised by regions experiencing either downwards and upwards flow directions. The regions experiencing downwards flow are shown with a negative

value in Figure 5. Regions along the Pathway of Investigation that experiences an upwards vertical flow component are shown having a positive value.

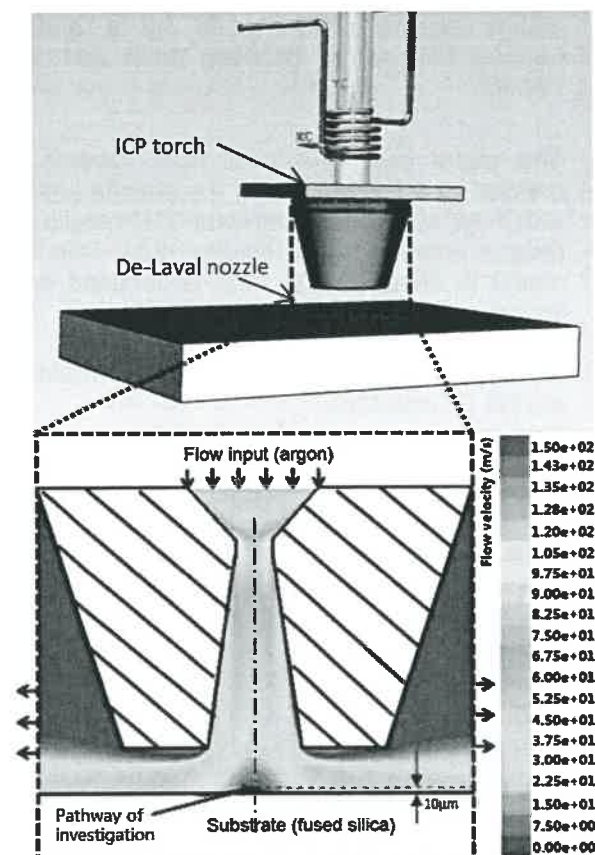


FIGURE 4. Overview of the CFD investigation. 3D drawing of the plasma figuring torch (upper); 2D CFD simulation illustration of flow velocity in the nozzle (lower).

The negative regions are considered to be those which will experience the presence of the radical compounds. The CFD model is able to determine the amplitude and direction of these particles and consequently help understand the regions of preferential etching.

From a processing view point, the plasma etching is considered to take place only in the region exposed to free radicals, because the fluorine radicals in the plasma jet can be in contact with the silicon atoms of the substrate. This assumption is supported by experimental tests performed by Dr Fanara. In the future this assumption will be further validated through tests using differing nozzle designs. However, there seems to be good initial indications supporting this assumption from footprint

experiment data carried out at Cranfield in 2010 by Dr Castelli. The 4 sigma value of the footprint 22 mm (11mm radius) is very close to the 10.75 mm radius exposed to free radicals as shown in Figure 5.

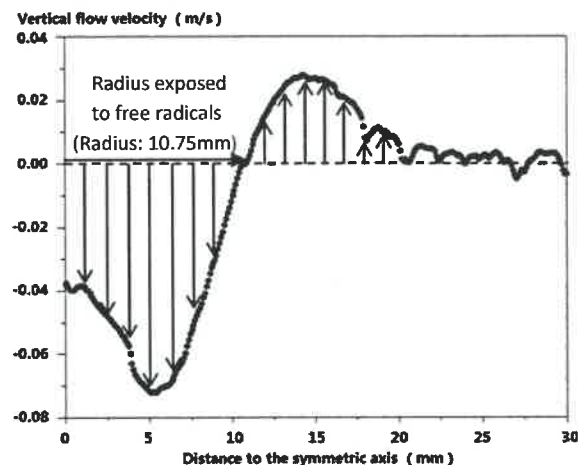


FIGURE 5. Vertical flow velocity plots along the pathway of investigation.

Figure 6 combines images from the gas flow simulation model and footprint experiment data. This figure highlights correlation between the material removal footprint and the regions along the "Pathway on Investigation" which experience a downwards vertical flow profile.

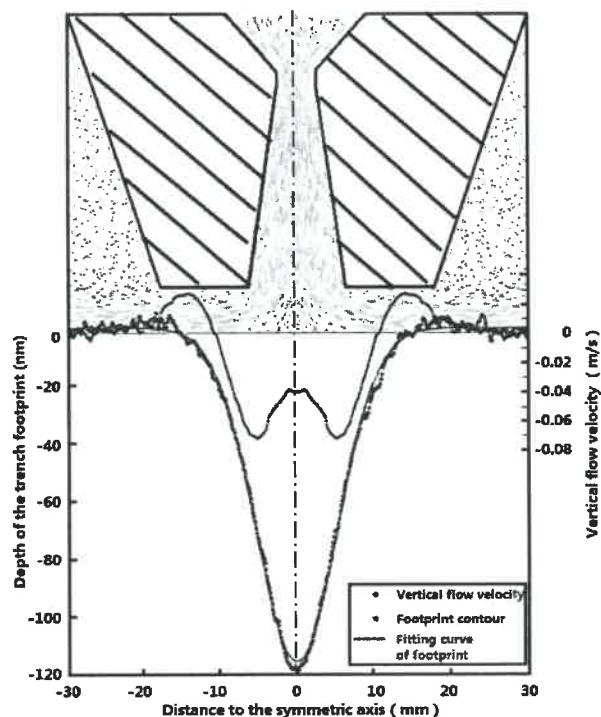


FIGURE 6. Curves of the etched area and gas velocity.

In Figure 6, the black curve shows the amplitude of the vertical velocity of the gas along the near surface regions of the substrate, whilst the blue plot shows the actual footprint removal profile obtained by Dr Castelli.

NOZZLE DESIGN EVALUATION USING CFD MODEL

Investigation of the nozzle's key design parameters using the aforementioned 2D axis-symmetric numerical model has been made. Evaluation is based on the changing radius along the Pathway of Investigation that is exposed to free radicals. The hydrodynamic characteristics of the De-Laval nozzle depend on seven parameters (shown in Table 1). Three of them have been changed to investigate their effect on gas flow. The three design parameters varied were: the diameter of the throat D_2 , the diameter of divergent end D_3 , and the depth of the divergent path h_3 . The following paragraphs detail the findings and results obtained through the CFD model focussing on the radius exposed to free radicals and the maximum velocity in the throat.

Effect of the Throat Diameter

The throat -parameter D_2 in the De-Laval nozzle is crucial, because the speed of flow increases strongly when the gas goes through this narrow section. A five-step increase of the throat diameter (4.0mm, 4.3mm, 4.7mm, 5.0mm and 5.3mm) was chosen for the investigation of the flow velocity change. The correlation between the diameter of the throat and the radius exposed to free radicals is highlighted in Figure 7. When D_2 decreases by 24%, the radius exposed to free radicals decreased by 19%. The sensitivity of D_2 is 0.79. Also it can be

observed that the flow velocity increases when the throat dimension gets smaller.

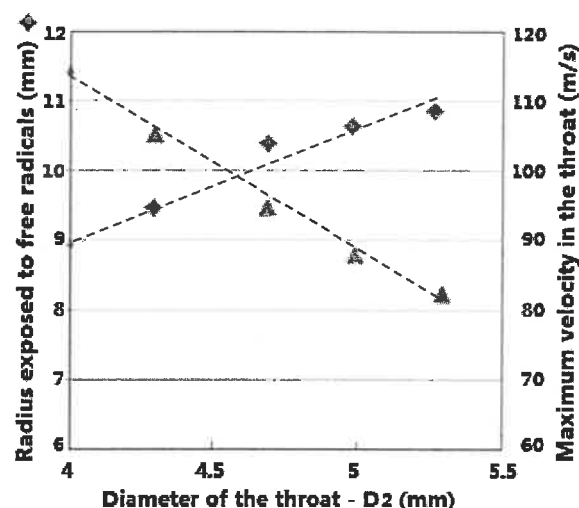


FIGURE 7. The throat diameter (D_2) versus the radius exposed to free radicals (♦ left), and D_2 versus the maximum velocity in the throat (▲ right).

Effect of the Divergent End Dimension

Similar comparison is made among nozzles with different dimensions of divergent end diameter - parameter D_3 . The chosen diameters were 10.8mm, 11.4mm, 12.0mm, 12.6mm, 13.2mm and 16.7mm. The correlation between the diameter of the divergent end and the radius exposed to free radicals can be seen in Figure 8. As expected, the results show that a wider divergent end generates a larger etched area. The dimension of the etched area decreases 8% as D_3 shrinks 18%. The sensitivity of D_3 is 0.44. The maximum flow velocity through the throat is little affected at these levels by this nozzle parameter.

TABLE 1. Parameters of the nozzle in the characteristic analysis.

No	Parameters	Symbol	Schematic plot
1	Diameter of convergent end	D_1	
2	Diameter of throat	D_2	
3	Diameter of divergent end	D_3	
4	Depth of convergent path	h_1	
5	Depth of throat	h_2	
6	Depth of divergent path	h_3	
7	Stand-off distance	h_4	

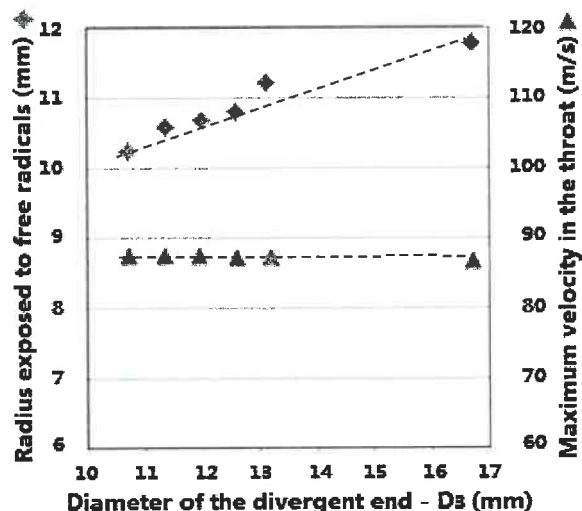


FIGURE 8. The divergent end dimensions (D_3) versus the radius exposed to free radicals (♦ left), and D_3 versus the maximum velocity in the throat (▲ right).

Effect of the Divergent Path Dimension

Through this third investigation, the influence of the dimension of the divergent path - parameter h_3 is analysed. Here two scenarios are investigated.

First scenario: the diameter of the divergent end is kept constant at 11.4mm. Thus the change of divergence path dimension affects the divergent angle which consequently increases. Unlike the last two comparisons, the obvious change of the divergent depth doesn't change the radius exposed to free radicals (Figure 9).

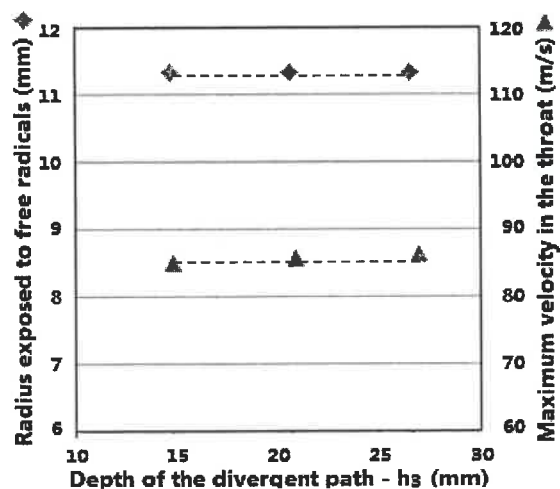


FIGURE 9. The depth of the divergent path (h_3) versus the radius exposed to free radicals (♦ left), and h_3 versus the maximum velocity in the throat (▲ right). (Divergent angle changes with h_3)

Second scenario: the angle of divergence is kept constant and therefore the divergent path dimension is altered. It can be seen in Figure 10 that the short nozzle reduces the radius exposed to free radicals. However, the response from h_3 is not obvious comparing to that of D_2 and D_3 .

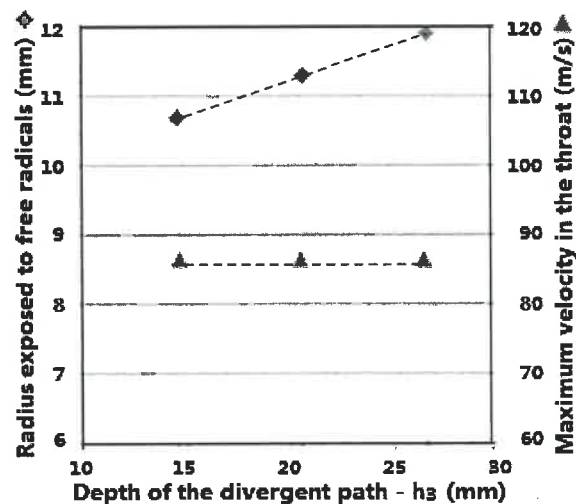


FIGURE 10. The depth of the divergent path (h_3) versus the radius exposed to free radicals (♦ left), and h_3 versus the maximum velocity in the throat (▲ right). (Divergent angle is kept constant)

These last two sets of results are complementary and they enable to correlate the radius exposed to free radicals and the diameter of the divergent end. Moreover, from the series of results obtained through this initial modelling work, there are three general design rules of the De-Laval nozzle:

1. Radius exposed to free radicals decreases significantly as the throat (D_2) shrinks;
2. Radius exposed to free radicals decreases when the divergent end (D_3) shrinks.
3. Smaller energy beam footprints should be achieved with adjustment of D_2 as it is more efficient than tuning D_3 .

CONCLUSIONS

An initial 2D axis-symmetry numerical model of an existing torch nozzle has been created. This simple model has indicated some sensible results when compared to actual process data of removal footprints. Correlation data gives confidence for more detailed modelling work.

Some initial design rules and nozzle parameter sensitivity analysis has been obtained. This

information can be used to create a number of new nozzle designs for future experiments.

Limitations of this CFD model include the use of argon flow instead of a plasma flow and insufficient supporting experimental data. These limitations will be addressed by the lead author during the next 2 years of his PhD.

FUTURE WORK

The CFD model will be advanced by:

1. Measurement using plasma diagnostic for more accurate parameters including initial entry flow velocity and temperature, heat loss in the nozzle;
2. Definition of the ionization and recombination of the argon particle and its electron so as to simulate argon plasma instead of hot argon pure argon;
3. Taking the turbulence and swirl into account;
4. Further validation through material removal footprint trials using differing nozzle designs.

Using the developed CFD model new nozzles and torch designs will be created. These new torch / nozzle designs will be employed to establish an effective plasma process that rapidly removes MSF for large scale metre class optics.

ACKNOWLEDGEMENT

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OPTIMAL VIBRATION ISOLATION FOR SHOCK-SENSITIVE MACHINES

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INTRODUCTION

The world is in motion. This fact is being taken for granted by most of us in daily life. But it can be a challenge for those working in precision engineering or related industries when it comes to problems caused by vibrations.

Vibrations can be caused by the machine itself or the surroundings, such as a nearby highway, railroad or other machinery. Today, we have to face drastically increasing demands regarding quality and precision in nearly all fields of the engineering process. Therefore, measuring devices cannot only be found in laboratories, but also closer to the production lines. This allows continuous monitoring of quality parameters as well as evaluation of selected product samples throughout the entire manufacturing process. Depending on the requirements regarding precision and accuracy, vibrations can be of great negative impact for the results of a production or measurement process. In general, the more precision is required, the more efficient the vibration control has to be.

Over the last few years, air spring systems in combination with high-quality mechanical level control have been the first choice of an optimal compromise between achievable accuracy and cost effort.

By using electronic valves, that enable precise and accurate adaption of the system control parameters, isolation efficiency and transfer function can be drastically improved. As mechanical and electronic level control systems can be combined with the same type of air spring elements, no constructive modifications at the machine bed or at the support of the isolators are required. Moreover costs of the entire isolation system do not increase disproportionately.

There are numerous advantages of air spring technology compared to steel springs, electromagnetic actuators or linear motors. The linear correlation between applied air pressure and resulting load capacity of air spring isolators

leads to a high flexibility and easy adaption for different load distributions.

Basic properties like natural frequency and dampening are nearly constant within a reasonable range of operation. Moreover air spring elements are characterized by a very high mechanical stability with low maintenance effort. They also do not require any additional damping elements. Their very low energy consumption is essential to avoid any heat generation and magnetic variation, which is essential for many applications, like e.g. electronic beam microscopes.

Fundamentals

The efficiency of any vibration isolation system significantly depends on the matching ratio η between excitation frequency and natural frequency of the isolator, see [1]. In general,

$$\eta = \frac{f_{\text{excitation}}}{f_{0,\text{isolator}}}$$

$$V = \sqrt{\frac{1+4D^2\eta^2}{(1-\eta^2)+4D^2\eta^2}}$$

the efficiency of vibration isolation increases as the natural frequency of the isolator drops. The graph in Figure 1 shows that the system only isolates when this ratio exceeds $\sqrt{2}$. If the ratio is less than $\sqrt{2}$, the vibration will be amplified due to resonance effects. The damping factor D determines the amplification resonance and the transmissibility with high frequencies.

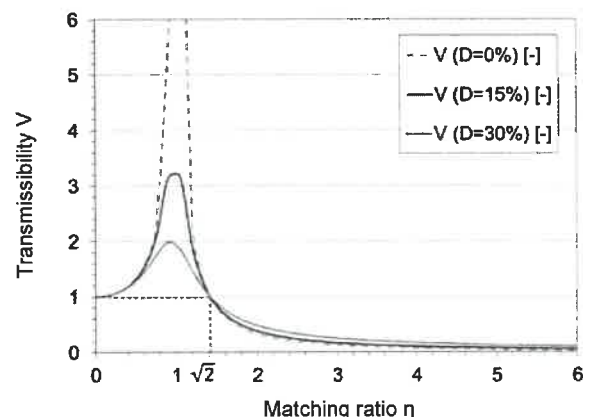


FIGURE 1. Transmissibility of a vibration isolator depending on damping and matching ratio

Typically, the objective is to achieve a ratio of between 3 and 4 [2], whereas a ratio of 3 is considered to be the minimum effective target value where an isolation efficiency of nearly 80% can be achieved, and a ratio of 4 to be an economic limit.

Electronic Pneumatic Position Control

Effective and cost-efficient solutions for nearly every problem caused by vibration – that is the claim Bilz Vibration Technology meets every day and very successfully worldwide. More than 45 years of experience in the field of vibration isolation have made the company based in southern Germany the European market leader today. Bilz offers a wide range of products for vibration isolation including: simple bonded rubber isolation pads, rubber and membrane air springs, mechanical and electronic level control systems, and both semi-active and active vibration cancellation systems with 3 or 6 Degrees of Freedom. This is why Bilz is able to provide an optimal isolation concept for nearly every application where vibration issues occur, starting from reducing shock emissions of forging hammers to the point of elastic machine support in the semiconductor industry.

The latest product by Bilz, the EPPC™ (Electronic Pneumatic Position Control), was launched in fall 2013. It is a real-time level control system which complements the Bilz product portfolio perfectly. The EPPC™ provides optimal vibration isolation for highly dynamic and shock-sensitive machines. Therefore, the EPPC™ can be installed with high-precision machines, where the focus is on optimal level accuracy of $\pm 8 \mu\text{m}$ over a 12 mm range and little deflection as well as short settling times at load changes. Therefore it can be used to isolate not only measuring devices and microscopes but also test equipment and production machines.

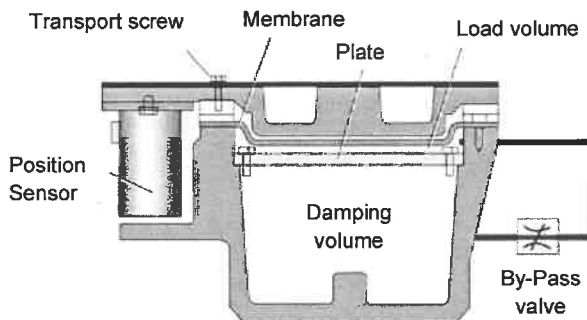


FIGURE 2. Air spring isolator BiAir® equipped with position sensor

The system is used in combination with standard BiAir® membrane air spring isolators, which can be mounted directly below machine supports, support platform or a massive foundation block. Figure 2 provides a schematic cross-sectional diagram of a BiAir® element.

The two chamber system design consists of a load and a damping volume. Both volumes are connected by an adjustable mechanical bypass throttle valve. Any deflection of the isolated machine on top of the element leads to a change of the size of the load volume and a resulting air flow from one air volume to the other through the bypass. Due to air friction inside the bypass, energy will be converted into heat and damping up to a maximum value of $D = 15\%$ can be created. The natural frequency of the given air springs ranges between approx. 1.1 and 2.5 Hz in vertical direction, whereas the horizontal direction is characterized by a natural frequency of 2.5 Hz.

Multiple air spring types and sizes are available to ensure that design and layout of the isolation system perfectly matches the individual demands of the specific application. In practice, air spring elements are typically designed to work with an air pressure between 2 and 5 bar and vary between 70 mm and 900 mm in diameter. This results in a total load capacity of less than 20 kg up to 15.5 tons per element.

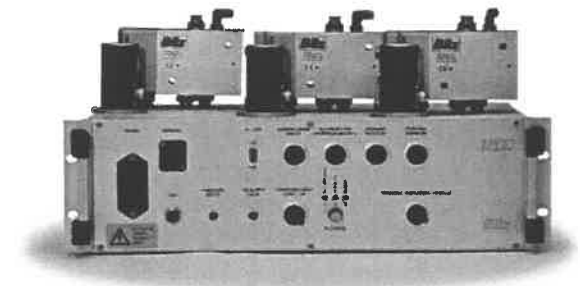


FIGURE 3. EPPC™ level control incl. position sensors and servo-valves (3 units each) as well as the electronic control unit

The EPPC™ can be combined with three (refer to Figure 3) or six groups of air springs to control up to 6 degrees of freedom. It monitors the machine position for every degree of freedom as well as internal air pressure of the air springs to control the dynamic behavior of the system. The performance of the passive air spring itself is significantly improved by using a high-performance 14bit AD converter and a 16bit PID

controller which allow nearly noise free regulation. Servo-valves are mounted very close to the air springs to eliminate control degradation through pressure losses in the tubes.

All these features enable the EPPC™ level control system to act as a semi-active system and create a very high damping factor up to 30% in the resonance frequency range. This leads to a significant reduction of the corresponding amplification or transmissibility factor.

In addition to the general advantages of air spring technology compared to systems driven by electro-magnetism (see section "Introduction"), the CAN-Bus topology allows a distance of up to 20 m (66 ft) from the isolation system to the electronic control unit itself. This way the isolation system can even be applied at highly sensitive locations and surroundings e.g. clean room environment and laboratories etc.

MEASUREMENTS AND PERFORMANCE

Figure 4 shows a measured disturbing frequency spectrum on the floor (Boden) and the resulting vibration frequency spectrum remaining on a Bilz EPPC™ isolated system (System).

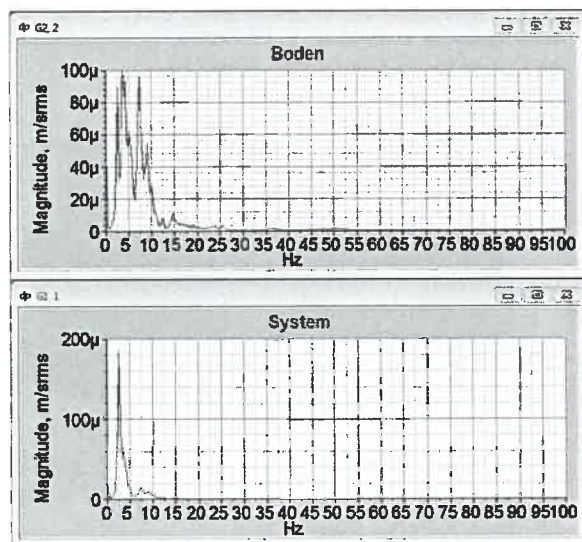


FIGURE 4. Frequency spectrum of vibration measurement with (System) and without (Boden) Bilz EPPC™ isolation system

The isolating air spring element, which is part of the given system, is characterized by a natural frequency of approx. 2.5 Hz in vertical direction. This leads to resonance amplification of approx. +100% within the $\sqrt{2}$ range around the natural frequency, see bottom diagram of Figure 4.

This value corresponds to the amplification factor given by the formal expression and its plot shown in Figure 1 in the "Fundamentals" section: The curve named "30%" represents the theoretical transmissibility function of the EPPC™ system. This system performance can be achieved when the maximum damping value of 30% is applied within the electronic system setup. In this case a maximum transmissibility value of 2 results in the resonance case ($\eta = 1$), which equals an amplification of 100%.

For frequencies higher than 3.5 Hz, the reduction of the vibration passing from the ground through the isolating system can be clearly observed. At 7.5 Hz ($\eta = 3$) a vibration amplitude of approx. 90 μm is reduced to nearly 15 μm which equals a vibration efficiency of around 85%.

In comparison to currently used mechanical pneumatic levelling systems (e.g. Bilz MPN) there is a significant increase in performance. The maximum damping factor of the mechanical systems of approx. 15% leads to a much higher resonance amplification, see again Figure 1. The settling time, which is required to reach and stay within a certain range, is drastically reduced, see Figure 5. For excitation amplitude of 80 μm , the response curve of the respective isolation system is shown. To reach a stable position within a range of e.g. 15 μm , the settling time reduces by 40% from approx. 1.25 s to 0.75 s.

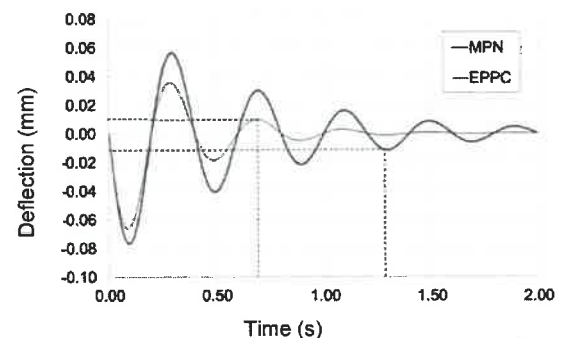


FIGURE 5. Settling time of electronic (EPPC™) and mechanical (MPN) pneumatic position control

For applications with very high dynamic loads due to rapid movement of work pieces or machine components like scanning units, tool changing units etc. with very high requirements for settling time and constant leveling, the performance of an EPPC™ and air spring based

isolation system can be further improved by adding additional mass to the system.

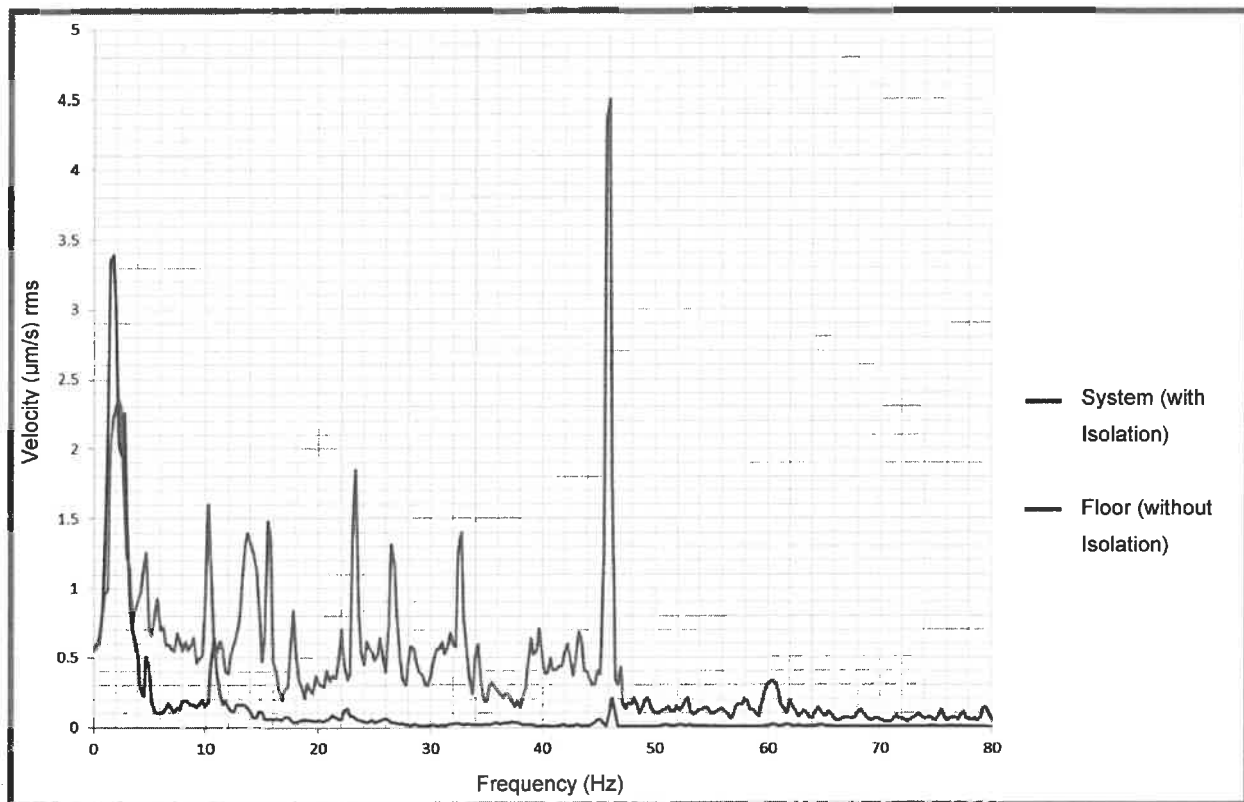


FIGURE 6. Vibration analysis of CMM with (System) and without (Floor) vibration isolation

Figure 6 shows the measured values obtained by a vibration analysis of an isolated coordinate measurement machine (CMM). In the described application, the footprint of the CMM is approx. 2.0 m x 5.5 m. The maximum height of the machine itself is more than 5.0 m. The weight of the CMM incl. work piece is more than 40 tons. High masses and large dimensions of machine components and work pieces desire a high quality isolation concept to ensure constant reproducibility of measurement results as well as optimal duration of the measurement process.

In the same time, the foundation block helps to avoid any torsion of the machine bed due to the elastic installation of the machine. Moreover the static layout of the system is improved by lowering the center of gravity of the complete setup.

Therefore, the isolation system was realized with the EPPC™ in combination with foundation isolation. In this example of an indirect isolation concept, a massive concrete block is put between the isolators and the isolated object. The concrete mass in the given project was around 100 tons.

In order to meet the requirements according to the limit values given by the CMM manufacturer and the vibration scenario coming from the environment from the CMM, the foundation block was put on Bilz BiAir®-HE air spring elements which are characterized by a reduced natural frequency of 1.7 Hz. The reduction is achieved by increasing the volume of the air springs.

Similar to the example discussed previously, the resonance amplification can be observed. The achievable high dampening of the EPPC™ results in a resonance amplification of less than factor 2. For excitation frequencies above 2 Hz, isolation efficiency is drastically improving with increasing frequencies.

CONCLUSION

The EPPC™ level control system provides highly effective vibration isolation in the field of precision engineering. The combination of high-performance electronic and pneumatic devices, optimized pneumatic design and technically mature air spring technology facilitates daily

work with optimal results even for critical applications.

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Characterizing Alternate Methods of Determining Velocity Information for Feedback Control of High Performance Stages

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INTRODUCTION

Feedback control loops for high performance stage positioning require not only input of stage position, but also velocity. Both must be accurate, have low noise, have good resolution, and must have low phase delay in order to obtain high performance. Modern position sensors can provide reasonable position information, but stage performance can often be limited by the quality of the velocity information, especially at low velocity. It is widely documented that simply differentiating position to create velocity creates a noisy velocity signal and that a high resolution position signal results better performance [1,2,3]. Various velocity estimator algorithms are used to create a higher quality velocity signal, but no single velocity estimator ranks best for all applications [4]. Velocity estimators always create phase delay, as a result of combining data taken over multiple samples, and the greater the number of samples, the greater the phase delay.

The purpose of this investigation is to evaluate alternate ways of achieving a high quality velocity signal. The methods we analyzed utilize features of a relatively new type of position encoder that achieves ultra high position resolution and also makes available an internally created ultra high resolution velocity signal that is not created by differentiating position [5]. Any model in Nikon's ModuRay[®] series of incremental position encoders has these required capabilities. For this investigation we used the LT700 model encoder head and scale, along with new and improved interpolation electronics that provide a bandwidth of 20kHz.

ModuRay linear encoders provide digital position resolution of 3.81pm, with 3- σ position noise that can be below 100pm. (For reference, the diameter of a single hydrogen atom is about 108pm.) The ModuRay encoders achieve such

high performance by modulating the wavelength of a VCSEL light source at 100kHz, which causes a diffraction pattern to move back and forth over the 4 μ m grating pitch. The position and direction information become embedded in the phase of a *single* modulated signal seen by the photo diode detectors. This modulated signal is decoded by high speed electronics, e.g., an FPGA, using a patented phase lock method. ModuRay encoders are able to achieve significantly low interpolation error and high signal-to-noise ratio compared to conventional A-quad-B output encoders. This is because the interpolation electronics are always decoding the phase signal of a single modulated AC signal, even when there is no stage motion, whereas conventional A-quad-B output encoders rely on the stability, noise, and accuracy of two DC levels for position calculation.

It is thought that differentiating a position signal with such low noise and quantization error will provide a low noise velocity signal. The noise and spectrum of this velocity signal are compared to the noise and spectrum of the internally generated velocity signal. Also, by removing the lowest 4 bits of position output we simulated the position output of an encoder whose resolution is not as good and computed the position noise and velocity noise, for comparison.

The primary concern of this investigation was the velocity signal resulting from a stage that is not moving or is moving at a relatively low velocity. The term "in-position stability" has been used to specify the case of a stage that is not moving. In-position stability is important in applications such as diamond turning machines in which one axis is supposed to be stationary or moves very slowly.

Some systems, such as scanning exposure systems, utilize a stage moving at a constant velocity. It is likely that a method of achieving a low noise velocity signal for zero or near zero motion will also be a critical requirement for precise and smooth motion in a constant velocity application.

THE INTERNAL VELOCITY SIGNAL

The interpolation electronics of ModuRay encoders contain a feedback loop which, among other things, creates a velocity output. Both the position and the velocity outputs are updated by a 40MHz clock, i.e., every 25ns. A diagram of the velocity generation section is shown in Figure 1. The filter is shown as a dotted block because the characteristics can be customized for a particular application.

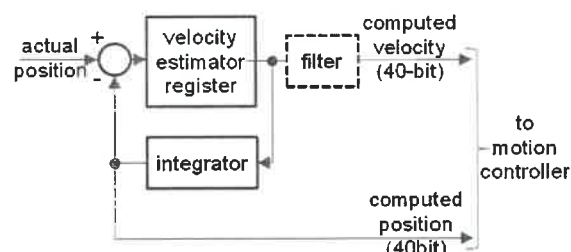


FIGURE 1. Block diagram of the basic signal processing flow in ModuRay interpolation electronics.

One LSB (Least Significant Bit) of the digital velocity signal employed for this investigation corresponds to $18\text{E-}15\text{m/s}$. (Such ultra high velocity resolution is probably unnecessary, and it is likely that we will determine that lower resolution, such as 1.1pm/s , will be more than adequate and will allow a higher maximum velocity using a 40-bit output.) The low-pass filter is included to provide anti-aliasing before the output signals are sampled by the motion controller, but this filter should probably be optimized to provide steeper attenuation at frequencies above the targeted bandwidth of the motion controller.

The phase delay of this velocity output, relative to the actual position of the stage, is determined by the transfer function of the loop, whose bandwidth can be as high as 20kHz for small amplitude stage motion. The data propagation latency is about 250ns, including the ADC pipeline delay. The optional anti-aliasing filter will also contribute phase delay in addition to the

20kHz bandwidth of the feedback loop for position and velocity decoding.

THE MEASUREMENTS

The measured velocity output is limited by the noise of the system, including both mechanical (including acoustic) noise and noise in the electronics, plus possibly noise due to quantization error. In an effort to achieve as little stage motion as possible, we created a rigid stage with no moving parts, as shown in Figure 2, and we measured the output when the stage was mounted on a high quality vibration isolation table.

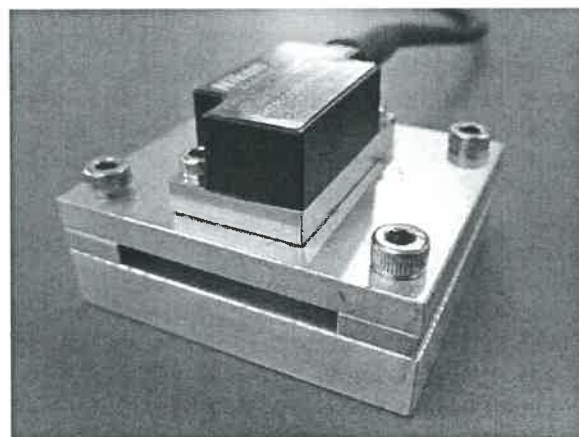


FIGURE 2. Immovable stage

We measured position and velocity, sampling at a rate of 20kHz and collected 100,000 samples (5 seconds). The standard deviation of position was 25.7pm, which means the $3\text{-}\sigma$ noise is well under 100pm. Figure 3 shows a typical FFT plot of the position output.

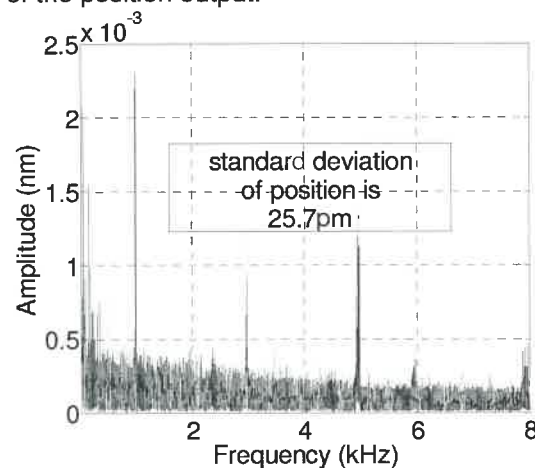


FIGURE 3. Typical FFT of Position output from a stage that is rigidly held in place.

The measurements shown above were taken in Japan. The peaks at 50Hz and 100Hz are probably due the AC line frequency, as similar measurements taken in the US showed peaks at the US AC line frequency of 60Hz and its harmonic. Commonly air currents, acoustic noise, and floor vibration caused by machinery can all cause mechanical noise. The vibration isolation table reduces the amplitude of floor vibration above 50-100Hz vibration by about 99%, but it doesn't eliminate it. The peaks at 1kHz, 3kHz, and 5kHz are probably due to acoustic noise, as these peaks moved when we made additional measurements.

Figure 4 shows the FFT of the internally generated velocity signal obtained during the same measurement as the position plot of Figure 3. In this case the anti-aliasing low-pass filter was a first order IIR filter with a pole at 100Hz. The standard deviation of velocity was 3.55nm/s and would be lower if we used a better anti-aliasing filter. (For reference, the hair on your head grows at a rate of about 5nm/s.)

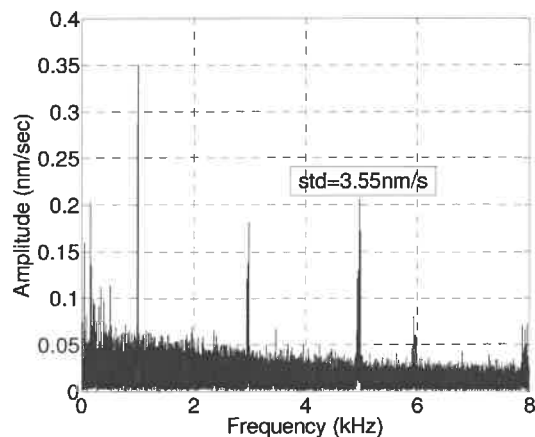


FIGURE 4. Typical FFT of the Velocity output of a stage that is rigidly held in place. This output is from the internally generated velocity signal.

The traditional problems with differentiating position to obtain velocity are caused by position resolution and a variety of issues such as measurement noise and interpolation error and quantization error. In ModuRay encoders, the position resolution is excellent, the position noise is very low, and the error due to quantization should be negligible. Therefore, for comparison, we differentiated the position output of the ModuRay encoder to create a velocity signal that we could compare to the internally generated velocity signal of Figure 4. Figure 5

shows that velocity signal, after applying the same 100Hz anti-aliasing filter used in Figure 4.

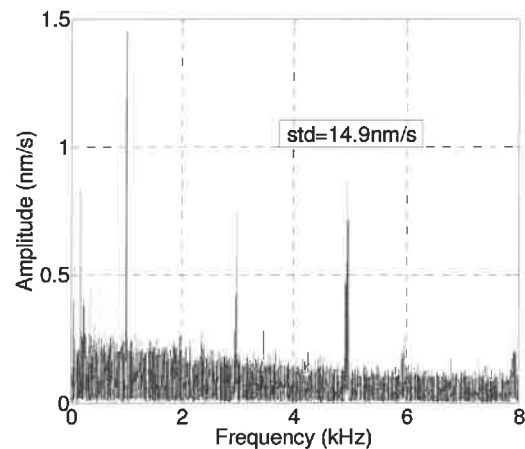


FIGURE 5. FFT of the Velocity signal created by differentiating the position output of a high resolution, low noise position encoder.

Figure 5 has the same shape as Figure 4, as one would expect, but velocity noise of the internally generated velocity signal is more than 4 times smaller.

Velocity estimators that average data over more than one sample are often used to reduce the noise caused by simple differentiation, at the expense of data latency because the velocity estimate is delayed by additional cycles of the motion controller sampling rate. In our case, we used a finite impulse response (FIR) filter of eight position samples because it was easy to implement. That velocity signal is shown in Figure 6.

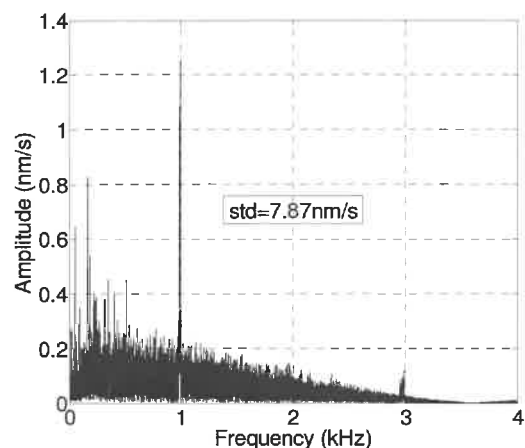


FIGURE 6. FFT of the Velocity signal created by a finite impulse response velocity estimator, using 8 position samples.

The shape of Figure 6 is the same as Figure 5 at low frequencies but drops off at higher frequencies, due to the FIR filter. The standard deviation of the velocity signal is a little lower but not nearly as low as obtained from the internally generated velocity signal. In the case of the FIR velocity estimator, the phase delay is increased, due to taking data from 8 samples, but it may still be dominated by a low-pass filter.

We also investigated the output of a simulated position encoder with lower resolution than the ModuRay encoder by truncating the four least significant bits of position output of the raw position data and then applying the same velocity estimator as used in Figure 6. Figure 7 shows the FFT of the velocity signal created by applying the same 8-sample FIR velocity estimator to the truncated position data. The velocity noise of 9.01nm/s is a little worse than the velocity noise obtained from the higher resolution output, indicating a benefit of ModuRay's resolution. The position noise is increased slightly from 25.7pm to 31.3pm.

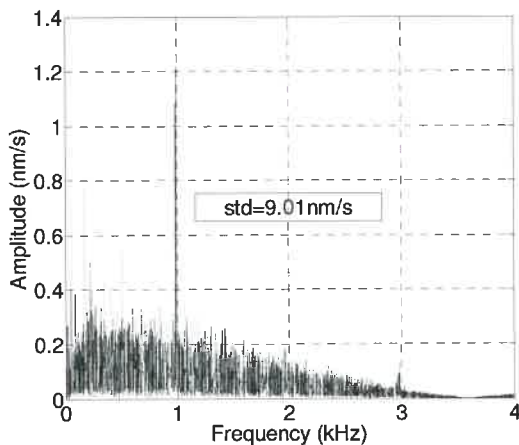


FIGURE 7. FFT of the Velocity signal created by a finite impulse response velocity estimator, using 8 truncated position samples.

CONCLUSIONS

Table 1 summarizes the results of the various velocity signals that we evaluated. The best velocity output is obtained from the internally generated ModuRay signal, as it has the lowest noise (standard deviation) and also the lowest phase delay. While different velocity estimator algorithms and different low-pass filters can be used, it is likely that this internally generated velocity signal will have the lowest noise and least phase delay in those comparisons.

TABLE 1. Summary comparison of velocity signals obtained via various means from ModuRay encoder.

	Std Deviation	Note
Internal @20KS/s	3.55nm/s	low noise high bandwidth
d(pos)/dt @20KS/s	14.9nm/s	higher noise, even with ultra pos resolution
Vel est @20KS/s	7.87nm/s	good noise but higher delay
Vel est(tr) @20KS/s	9.01nm/s	higher noise using lower resolution position data

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LINEAR MICRO-ACTUATION SYSTEM FOR PATCH-CLAMP RECORDING

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ABSTRACT

Measuring the electrical activity of neurons is essential for understanding how they encode and transmit information in the brain. Using a technique known as patch-clamping, the electrical activity of single neurons can be reliably recorded by pressing a small glass pipette filled with electrically conductive and pneumatically controlled solution against the neuron's membrane. This requires accurate and repeatable mechanical control of pipette position, typically necessitating a bulky actuation system and thus making it difficult to position several pipettes around a tissue specimen to record from multiple neurons at once. We have developed a linear micro-actuation system for patch-clamping that exhibits high positional accuracy ($< 150 \mu\text{m}$ on-axis error over full travel), high repeatability

(on-axis $\sigma = 33 \mu\text{m}$ for full travel; $\sigma = 0.71 \mu\text{m}$ for $15 \mu\text{m}$ travel) and low drift ($0.61 \mu\text{m}/\text{hour}$). The system was designed and fabricated to patch-clamp onto neurons in a mouse brain slice. The miniaturized device presented here makes it possible to position up to 21 actuators around a $5 \times 5 \text{ mm}$ tissue sample and eventually record intracellularly from a large number of neurons.

INTRODUCTION

Neurons are electrically active due to the controlled flow of ions through pores in their membranes. Patch-clamping, the gold standard technique for measuring trans-membrane voltages and currents, involves delicately resting a $1 \mu\text{m}$ diameter pipette against a cell to create

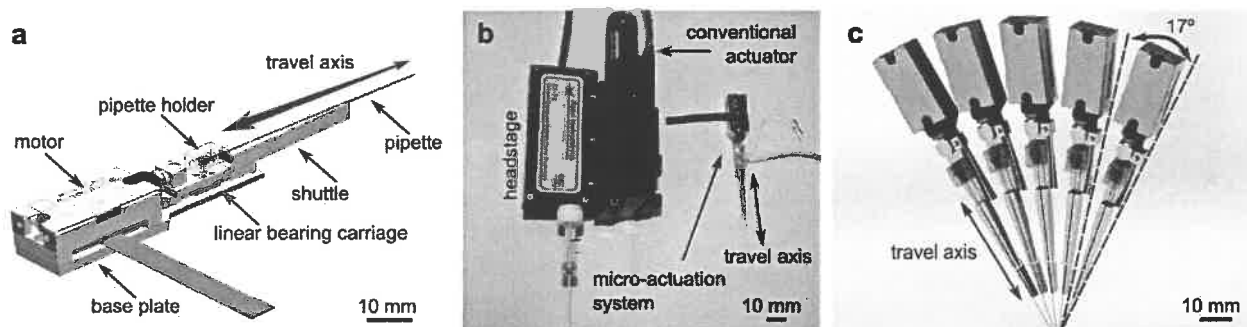


FIGURE 1. Micro-actuation system for patch-clamp recording. (a) CAD model. (b) Conventional actuator, pipette holder and headstage [left] and our micro-actuation system [right]. (c) Proposed CAD design of micro-actuation array. The small ($\sim 17^\circ$) angular profile a single micro-actuation system can enable high-channel recording from small ($\sim 5 \text{ mm} \times 5 \text{ mm}$) tissue specimens (not shown).

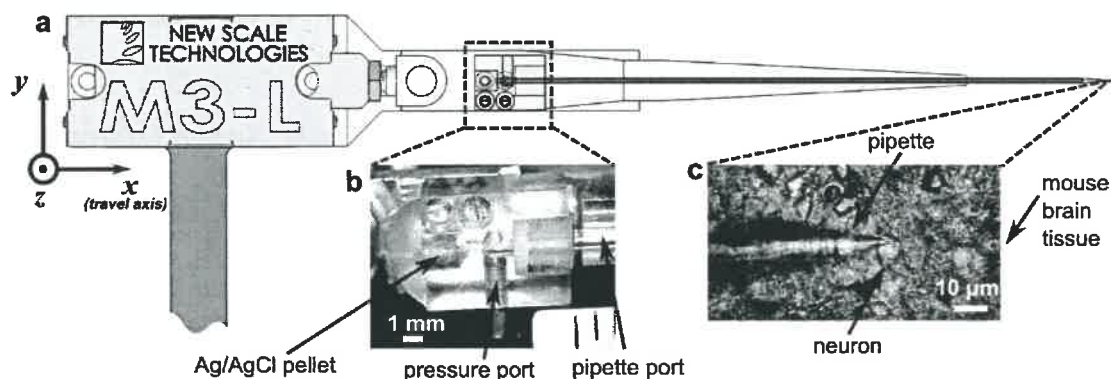


FIGURE 2. Detailed design and testing of the linear micro-actuator for patch-clamp recording. (a) Micro-actuation system based on the M3-L Linear Actuator (New Scale Technologies; Victor, NY, USA) (b) Micro-machined pipette holder. (c) Differential interference contrast (DIC) image of a patch-clamping experiment. A neuron in a mouse brain slice has been patch-clamped using a pipette mounted on the linear micro-actuation system.

an intimate electrical and mechanical connection between the pipette tip and the neuronal membrane. One can then apply suction to 'break-in' to the neuron to attain a 'whole-cell' configuration in which the contents of the cell are directly accessible [1]. From there, it is possible to record single-neuron electrical activity with a very high signal-to-noise ratio. Different features of this activity are important for elucidating brain function – for instance, high-amplitude "spikes" of the membrane voltage are the primary method of inter-neuronal communication in the nervous system.

The size of conventional pipette micro-actuators is the current limiting factor when attempting to record from many neurons simultaneously in a small region of interest in the brain. Commercial systems have been outfitted to support up to 12 independent pipette actuators [2] but due to their large size, this is near the physical limit.

Here we present a novel linear micro-actuation system (Fig. 1a) that exhibits mechanical characteristics sufficient to perform patch-clamping. Our system has a volume that is less than 1% and a weight that is only 1.4% of a conventional actuator and holder (Fig. 1b). This miniaturization of actuation systems will enable the use of many closely packed electrodes (Fig. 1c) that can simultaneously access many neurons, as is desirable for high-channel probing of neural circuits [3].

MATERIALS AND METHODS

Detailed design

The linear micro-actuation system in Fig. 2a is based on the M3-L motor (NewScale Technologies; Victor, NY). The motor has a travel range of 6 mm and a motion resolution of 500 nm. The motor pushes a custom-made aluminum shuttle mounted on a miniature linear ball bearing carriage. The carriage rides on a 60 mm long guide rail (McMaster-Carr 8381k27).

A custom pipette holder (Fig. 2b) was machined out of cast acrylic and mounted on the shuttle. Three ports were drilled in the pipette holder: (1) a port to house an Ag/AgCl pellet for electrical connectivity with the pipette, (2) a pressure port to connect to an air pressure controller, and (3) a port to constrain the pipette. The three ports are interconnected and are filled with physiological saline to ensure electrical connectivity between the Ag/AgCl pellet and the pipette. The guide rail and the motor are mounted on a custom-made base plate. The base plate and shuttle were machined out of 6061 aluminum plate using a HAAS OM-1A CNC office mill. The pipette is pulled from 1 mm OD / 0.5 mm ID capillary glass using a Flaming/Brown P-97 micropipette puller (Sutter Instrument Company; Novato, CA) to a ~ 1 μ m diameter tip (Fig. 2c).

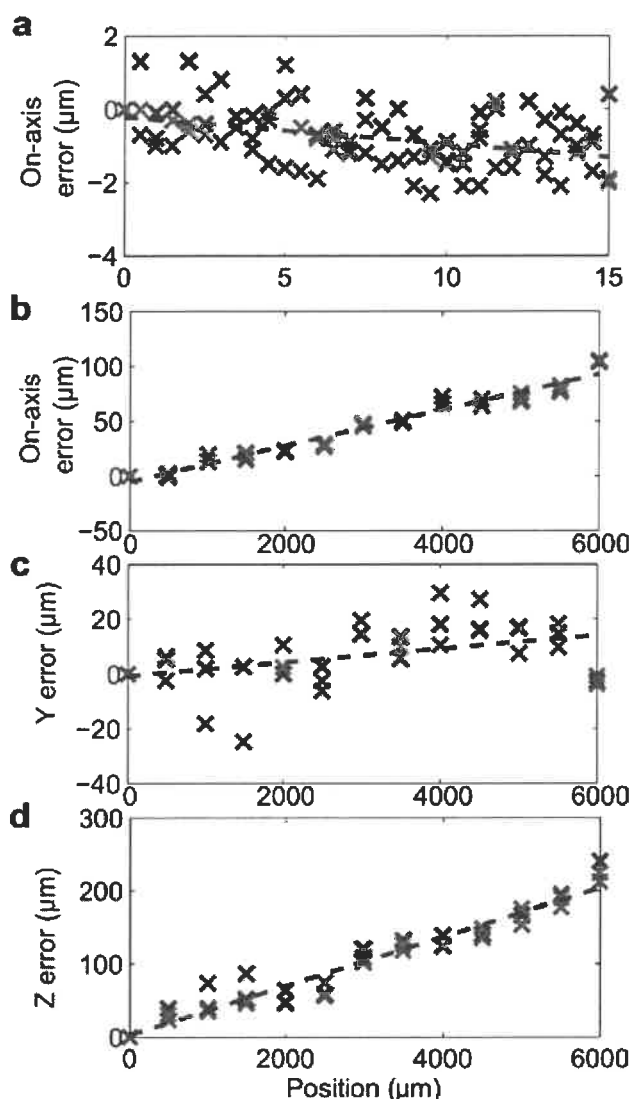


FIGURE 3. Accuracy measurement. (a) On-axis accuracy over a 15 μm travel distance. (b) On-axis and (c), (d) off-axis accuracy over full travel 6 mm. Three trials (black, blue, red) were performed. The error in motion was nearly linear (dashed line).

Benchtop testing

The accuracy, repeatability and drift of the micro-actuation system was evaluated. A pipette was fit into the pipette holder and clamped to the shuttle with an o-ring. The motor was mounted flat on a motorized XY stage (resolution = 100 nm) under a microscope equipped with a motorized 40x objective (UPLFLN 40X, NA = 0.75, Olympus;

Center Valley, PA) and differential interference contrast (DIC) optics (Scientifica Ltd, East Sussex, United Kingdom). This metrology system has a resolution of approximately 500 nm. Pipette position in three dimensions was determined by reading out motor encoder values of the XY stage and the objective when the pipette tip is centered and focused over a stationary on-screen crosshair.

Actuator accuracy is important when it is necessary to target specific structures in the brain. For instance, layers of the mouse cortex are tens to hundreds of micrometers deep in the brain [4], making positioning within $\sim 10 \mu\text{m}$ necessary for reliably targeting them. To test accuracy, the motor was first moved over 15 μm in 500 nm increments and subsequently over its full range of travel (6 mm) in 500 μm increments. The movement error was the difference between the actual pipette position and the commanded position as measured using the XY stage encoder averaged over $n = 3$ trials.

Repeatability of actuator motion is vital for algorithmic finding of cells based on electrical resistance [5]. It is also important for the ability to repeatedly target a brain structure for patch-clamping. The repeatability of motion was evaluated by commanding the motor to approach the middle of its travel (3 mm mark) from both directions in 1 mm increments. It was quantified by computing the standard deviation (σ) of the differences between the commanded position and actual position of the pipette tip over $n = 30$ trials.

Positional drift of the pipette tip after a whole-cell configuration has been established is detrimental to the stability of the membrane-pipette interface and ultimately worsens the signal-to-noise ratio of the intracellular recording. Therefore a system with minimal drift ($< 1 \mu\text{m}/\text{hour}$) is required. Motor drift was measured with the pipette acting as the load. Measurements were taken every 30 minutes for 2.5 hours.

In-vitro testing

We performed patch-clamp experiments in mouse brain slices [6] under DIC optics (Fig. 2c) by mounting the micro-actuation system on a

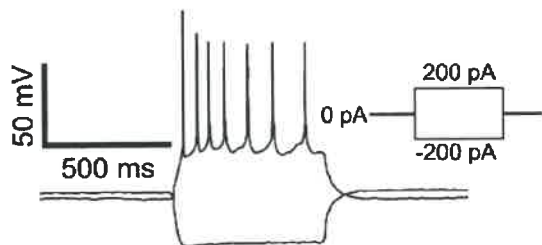


FIGURE 4. *Electrical activity recorded from the neuron shown in Fig. 2c in response to positive and negative current stimulation (inset) includes action potentials (large positive spikes) as well as passive resistive/capacitive components, both of which are typical electrophysiological characteristics of a patch-clamped cell.*

three-axis manipulator. Electrical signals were recorded using a standard patch-clamp amplifier and data acquisition software (Molecular Devices; Sunnyvale, CA). Power to the motor was disconnected during the recording to minimize electrical interference.

RESULTS AND DISCUSSION

Mechanical performance

In benchtop testing, the micro-actuation system exhibited on-axis accuracy down to 2 μm in small movements and 150 μm in full-travel movements (Fig. 3a,b). The large full-travel on-axis error was systematic and linear ($R^2 = 0.96$) and can therefore be predicted and compensated with a linear slope correction of $m = 0.01639 \mu\text{m}$ adjustment per 1 μm of on-axis travel to satisfy the 10 μm accuracy requirement for patch-clamping.

The off-axis accuracy of the system reached 14 μm in the y direction and 93 μm in the z direction (Fig. 3c,d). This is insufficient for targeting specific areas in the brain so additional motion constraints will need to be placed if highly accurate motion is needed.

After linearization, the repeatability of small actuator displacements (up to 15 μm) is $\sigma = 0.71 \mu\text{m}$ which is sufficient for algorithmic finding of cells based on electrical resistance. The on-axis repeatability over the full range of travel was

within $\sigma = 33.1 \mu\text{m}$. This is sufficient for repeatable targeting of brain structures such as distinct neuronal layers in the cortex of a mouse. The off-axis repeatabilities over the full range of travel were within $\sigma = 203 \mu\text{m}$ and $\sigma = 154 \mu\text{m}$ in y and z directions, respectively.

The pipette mounted on the micro-actuation system drifted a total distance of 4.22 μm over 4 hours. The average drift was 0.61 $\mu\text{m}/\text{hour}$. Based on average recording times of ~ 1 hour, this drift is negligible.

In-vitro experiment

We tested the usability of a single micro-actuation system by performing patch-clamp experiments in mouse brain slices. Representative intracellular electrophysiology traces (Fig. 4) are qualitatively indistinguishable from those obtained using commercially available pipette holders and actuators.

CONCLUSION

We have designed, manufactured, and tested a linear micro-actuation system that has sufficient mechanical characteristics for performing patch-clamp experiments. It was subsequently validated in an in-vitro patch-clamp experiment.

The system presented here uses many off-the-shelf components and can therefore be easily replicated in order to increase the number of signals that can be simultaneously obtained from brain tissue. The angular profile of the entire assembly is only 17° , which potentially allows for a circular arrangement of, conceivably, up to 21 single-axis actuators around a tissue sample. This would theoretically enable simultaneous recording of up to 21 intracellular signals, and eventually more after additional miniaturization. These technical advances can open the door for performing simultaneous high signal-to-noise recordings from larger groups of neurons than previously possible.

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EFFECTIVE CALIBRATION AND IMPLEMENTATION OF GALVANOMETER SCANNERS AS APPLIED TO DIRECT METAL LASER SINTERING

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INTRODUCTION

Direct metal laser sintering (DMLS) is the most common process by which metal parts are presently being produced through an additive manufacturing approach. These laser processing systems generally utilize 2-axis galvanometer scanners as a means of manipulating the planar position of a laser spot to produce sintering of the powder bed. The dynamic positioning performance of the laser spot directly impacts the dimensional accuracy of each sintered layer and the final part dimensions. As the scanner is the primary means of creating relative motion between the laser spot and powder bed, calibration and characterization of the scanner's ability to position the laser spot is the best way of improving the dimensional accuracy of the finished sintered part.

Current methods used in the calibration of two-axis galvanometer laser scanners include several novel techniques, but the most effective and readily validated method involves the measuring of a part marked by the system. The "mark-and-measure" approach is the most reliable of current calibration methods because the output of the given process, a part marked in-situ, is the artifact onto which metrology is carried out.

This paper summarizes the findings of a case study on effective means of 2-D galvanometer scanner calibration performed at Aerotech Inc. The study explored the use of 2-D laser scanners used in combination with field flattening optics, F-Theta lenses, and the findings and results are not necessarily applicable to other laser scanner arrangements. The study was able to produce micron-level

static laser spot positioning accuracy over a large majority of the scanner's field of view.

MARK & MEASURE CALIBRATION

The mark-and-measure calibration technique is widely used in laser processing systems. The general procedure is to mark patterns on a material, measure the pattern in the global space, and to use the difference between each mark's desired and measured location as a correction factor in future commanded moves. In the specific procedure followed during the case study described herein, two separate systems were used. One was a marking station fitted with a particular Aerotech scanner to imitate its integration into a laser processing system, and the second was a separate machine vision inspection station used to measure the parts produced by the marking station. Both systems are shown below in Figure 1. Grid patterns were burned into substrates at the marking station, and then the substrates were moved to the machine vision system for inspection. The data taken by the inspection station were appropriately post processed and turned into a two dimensional error array for the purpose of implementation into a calibration file in the scanner's controller. A subsequent substrate was marked and measured to determine the effectiveness of the calibration file. This process was repeated iteratively to produce the best results.

Several factors associated with the consistency of the spatial relationship between the scanner, the laser, and the substrate, as well as aspects related to the interaction of the laser and the substrate during the marking process were found to affect the success of a scanner's calibration via this mark-and-measure methodology.

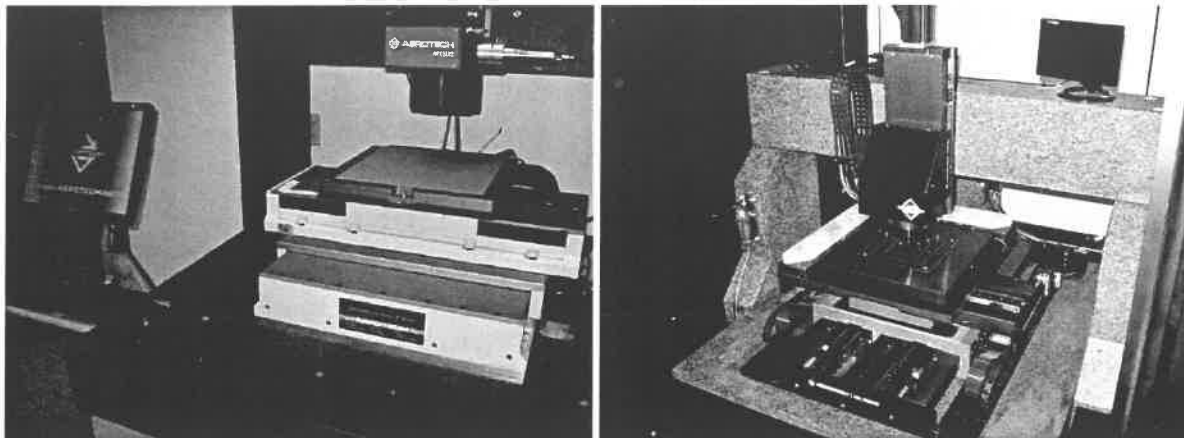


FIGURE 1. Images of the Laser Marking Station (left) and the Machine Vision Inspection Station (right) used in the laser scanner calibration case study.

ERROR SENSITIVITIES IN SCANNER SYSTEMS

The accurate and precise marking of substrates was found to be affected by several factors involving the physical setup of the marking station. Marking field to focal plane parallelism, working (marking) height consistency, process flatness, and input beam alignment consistency all influenced the calibration, resultant accuracy, and marking consistency of the marking station. Unfortunately, these factors are not mutually exclusive, and their independent effects on the marking process are not easily differentiated, and often combine. As a result, each is specifically defined and its impacts on the marking process described below, in order to clarify their individual importance to the calibration and implementation of scanner units.

All effects of the error sensitivities discussed in this section vary in severity with the choice of F-Theta lens, focal length of the system, and field of view, but all can also be readily corrected through a mark-and-measure calibration procedure, as long as the aspects of the system's alignment which produce them remain constant. Therefore, the major concern regarding accurate and precise laser marking is the consistency of each spatial relationship discussed below. Deviation in any of the below alignments from the state in which the scanner was calibrated will invalidate the calibration of the marking system to some extent.

Marking Field & Focal Plane Parallelism

As used in this document, the "marking field" is defined as the planar surface on which the laser beam is intended to mark; it is bounded by the geometry of the part to be marked. Additionally, the "focal plane," as used herein, is defined as the virtual plane over which the optical elements

of the system focus the laser beam; it is bounded by the available scan area of the particular scan head/lens combination in use. The parallelism of these two planar surfaces is defined here as the magnitude of the angle between the normal vectors of each plane, and when referring to the effects of this parallelism, it is assumed that the centroid of each plane is coincident with one another and that both are perfectly flat. Through testing, the alignment of the two was found to have an impact on the marking accuracy, and therefore, the calibration of the scanner. As the input beam is directed through travel by the scan head, it inherently has an incident angle greater than zero as it falls on the focal plane (except for where the beam enters the lens coincident, and parallel to the lens' axis.) Any misalignment between the marking field and focal plane causes the beam to be projected a distance that is different than the intended working height, and, as a result of the non-zero incident angle at the focal plane, a mark placement error is induced on the marking field. The total (pk-pk) induced error over the marking field from a parallelism error is linearly dependent on the total scan length, and is non-linearly affected by the nominal incident angle of the laser beam on the part and the parallelism error between the marking field and focal plane. In instances where a non-telecentric focusing optic is used (non-zero nominal incident angles,) the peak-to-peak induced laser placement error can be on the order of microns per 100 micro-radians of parallelism error.

Marking Height Consistency & Process Flatness

Similar to errors caused by marking field and focal plane misalignment, errors can be induced when the marking field is placed at a different distance from the lens along its axis than the

theoretical location of the focal plane. This height differential can either be a result of a constant offset between the marking field and focal plane caused by a lack of consistency in the parts vertical placement in the system or be a result of the parts flatness with respect to the focal plane or a combination of the two. Errors induced by this height variation are solely a result of non-zero incident angle of the laser beam at the focal plane. Therefore, the use of telecentric focusing optics effectively eliminates this error source; however, if non-telecentric optics are used, induced peak-to-peak laser placement error across the scan length can easily be on the order of microns per five microns of marking field height variation. This placement error is linearly affected by variation in marking field height, and is dependent on the beam's incident angle at a given position within the focal plane.

Laser Alignment Consistency (Zero Offset)

The final error sensitivity of major concern when calibrating laser scanners is that associated with laser alignment. The largest part of the laser placement error to be corrected in scanner systems is injected from the focusing optic's distortion of the focal plane's coordinate frame produced from field flattening. The exact manner in which this distortion is manifested over the focal plane depends on the boundaries of the laser beam's entrance into the lens. The lens' distortion of the beam's projected position on the focal plane is a function of the incoming beam's distance from the axial center of the rear lens objective. The center point of the focal plane, or "zero" location of commanded moves, ideally corresponds with the laser beam passing coincident and parallel to the lens' axis. In this ideal condition, the theoretical lens distortion of the focal plane has two axes of symmetry, which coincide with the (X) and (Y) axes of the focal plane, as defined by the scanners' motion.

Realistically, the beam incoming to the scanner is never perfectly aligned, and as a result, the laser beam being projected to the commanded center point of the focal plane is linearly offset from the axial center of the lens as it enters the rear objective. This is what's referred to herein as a "zero offset." Due to this zero offset, the actual lens distortion imparted on the focal plane is asymmetrical.

Any change in either the linear or angular alignment of the input beam to the scan head will induce significant laser placement errors, in this case, at the focal plane itself (independent of the previously mentioned error sensitivities.)

The level of this induced error is significantly more sensitive to angular changes in the alignment of the input beam. Angular changes in the input beam are projected through the scan head over a long enough distance that even a small change can create a substantial change in zero offset at the lens' rear objective.

The manner in which a change in the zero offset of a scanner system induces error is through the shift between the location of the lens and the incoming laser beam, and the associated shift in the focal plane's distortion. For example, when applying a correction table via a calibration file, each value in the correction table is associated with a commanded position in the focal plane. However, the magnitude of each value is associated with where the laser beam passes through the lens relative to its axis when at its associated command position. Therefore, when a change in zero offset occurs, there is a relative shift between the commanded locations in the focal plane, and the location of the laser beam with respect to the lens. This shift destroys the validity of the connection between the magnitude of a correction in the table and the location it's associated with. As depicted in Figure 2, a zero offset shift from (X0, Y0) to (X'0, Y'0) changes every position of the laser beam throughout the focal plane the same amount relative to the lens' axis, thereby changing the magnitude by which the lens distorts each commanded position's global location on the focal plane. This relative shift between the focal plane and the lens' distortion map is depicted by the shift in (X1, Y1) to (X'1, Y'1) and (X2, Y2) to (X'2, Y'2). As a result of this shift, the correction counts intended to correct the lens' distortion at (X1, Y1) over correct for the distortion that exists at (X'1, Y'1) because it is now closer to the lens' axis and is distorted a lesser amount. Likewise, the correction counts intended to correct the lens' distortion at (X2, Y2) are now not enough to fully correct the distortion seen by (X'2, Y'2) because it is farther from the lens' axis and is thereby further distorted. Therefore, the amount of error induced by a change in input beam alignment is dependent on the magnitude and type of the change, the size of the AGV unit in use, and the slope of the correction table being employed by the calibration file (which is primarily a factor of the F-Theta lens in use.)

Lastly, while the effects of even large zero offsets, when left unchanged, can be effectively calibrated, their impact on spot distortion cannot. A shift in zero offset also changes the distortion imparted focused laser spot.

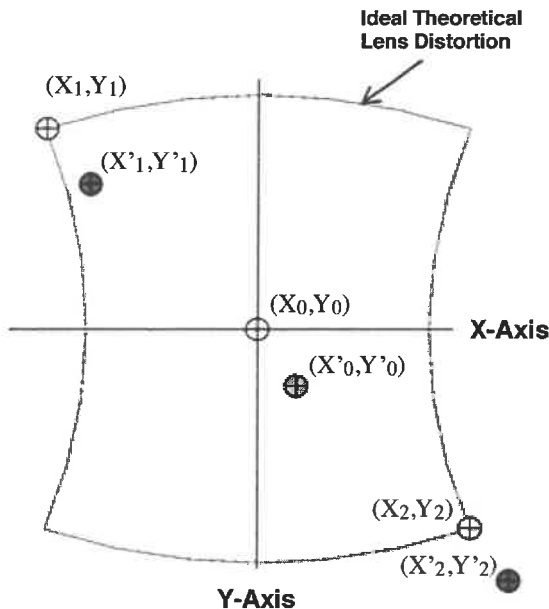


FIGURE 2. Depiction of a relative shift between commanded locations in the focal plane and a map of the ideal theoretical distortion created by an F-Theta field flattening lens, referred to as a change in zero offset.

Lens manufacturers offer theoretical software simulations of spot size deviation and focus quality as a function of position within the focal plane. As a result of non-ideal laser alignment, spot size deviation will be larger and patterned differently than what is predicted by the manufacturer. This can have a drastic effect on the quality of marks used for calibration as well as weld quality and stability related to where the laser spot is in the focal plane while performing a sintering operation.

SUBSTRATE VIABILITY & MARK QUALITY

As a result of good spatial control of the laser marking station studied, uncertainty in the measurement of calibration artifacts was found to be the primary limitation to the scanner's calibration. Measurement uncertainty and therefore the effectiveness of a calibration are dependent on the substrate as well as the quality of marks being made on that substrate when used in the mark-and-measure procedure outlined above. As previously mentioned, the calibration in the study employed the use of a machine vision system. As such, the level to which the mark-and-measure procedure was able to calibrate a scan head was directly affected by the camera's ability to accurately measure each mark in the grid patterns created. The marked substrate needed to produce a high level of contrast in the camera image; it needed to have exceptional surface finish to provide a solid background; lastly, the marks needed to be

thin and had to have crisp edges in order to produce the highly repeatable marking and measuring necessary for high accuracy calibration.

Measurement Repeatability & Marking Repeatability

Two major limiting factors in scanner calibration are the consistency with which the camera determines the center of a given mark, and the consistency with which the laser makes marks on the substrate. The nomenclature coined to represent these two factors is "measurement repeatability" and "marking repeatability." Measurement repeatability is defined herein as the maximum peak-to-peak difference in the measurement of a single mark's center between subsequent measurements of the same grid of marks. Conversely, marking repeatability is defined as the maximum peak-to-peak difference in the measurement of subsequently marked grids, assuming an identical setup and marking field to focal plane relationship. As such, marking repeatability can never be exclusively determined because it is inherently inclusive of measurement repeatability and the effects of the aforementioned error sensitivities. It must be estimated. An illustration of measurement and marking repeatability is provided in Figure 3. Combined, the measurement repeatability and marking repeatability establish the effective resolution of the calibration, and, therefore, must be minimized to obtain high levels of marking accuracy.

Making High Quality Laser Marks

Through the case study, it was found that thin marks produce the best results in both measurement and marking repeatability. Thinner marks produce better results because they fundamentally do the least amount of damage to the substrate, and thereby impart the least amount of heat into the substrate. This causes very thin marks to have minimal heat affected areas and insignificant mark growth relative to the theoretical focused spot size. Mark growth due to excess heat and/or a large heat affected area are major causes of mark randomness in addition to the substrate surface consistency. Minimizing excess heat input is the key to increasing the repeatability of a mark's shape (marking repeatability.)

Also, minimizing damage and heat input makes the contrast transition band between marked area and background very narrow in the camera image. The abrupt contrast change is what is described here as "crisp" marking. Having very crisp marks provides fewer inputs to the

camera's edge search algorithms, reducing the total number of influences on its opinion of a mark's center. Limiting the amount of pixels involved in the camera's search algorithm, therefore, gives it fewer items to change its opinion over, and showed to improve measurement repeatability substantially.

The ability to make the small, crisp, high quality marks described above is dependent on two factors, substrate selection and the laser parameters used for marking. Both are important to achieving high quality marks, but the ideal laser marking parameters for making a high quality mark can differ from substrate to substrate, making them difficult to specifically define.

As mentioned above, high accuracy calibration using machine vision requires that the substrate have superb surface finish, produce crisp marks with good contrast, and be strictly dimensionally controlled. As such, it was found that glass substrates with optical quality thin film coatings made good substrate candidates. However, using a coated substrate required certain considerations in the selection of laser parameters. When a laser has to burn through any thickness of coating to make a mark readable, it will impart more energy to the surface layer than is required to do damage as the beam passes through the coating's thickness. This naturally causes an increase in heat affected area and mark growth at the surface layer, which, in this case, was the camera's image plane. Not only is the mark,

therefore, inherently bigger in the image plane than the laser spot, but also it inherits randomness in its size, edge, and overall shape. All three reduce overall calibration resolution (marking & measuring repeatability.) It is imperative to impart as little thermal energy as possible to the area of surface layer surrounding the laser spot while fully burning or ablating the area underneath the spot to achieve quality marks. These thermal effects, mark growth and heat affected area, are not unique to marks made on coated substrates; they are only magnified by the coating. High accuracy mark-and-measure calibration demands that these effects be minimized regardless of the materials and processes used for marking. This minimization is achieved through the selection of the laser marking parameters.

Typically, the operator of a laser marking system has control of certain explicit laser marking parameters, such as average output power or pulse energy, pulse width or modulation frequency, marking speed, etc. However, it is the implicit marking parameters that determine the overall quality of a mark, specifically average power density, pulse power density, and average power seen by the substrate. The key to making quality marks that approach the theoretical laser spot size without producing the undesired thermal effects is to achieve high average and pulse power densities while reducing the average power dumped into the substrate. Examples of poor and high quality laser marks are shown below in Figure 3.

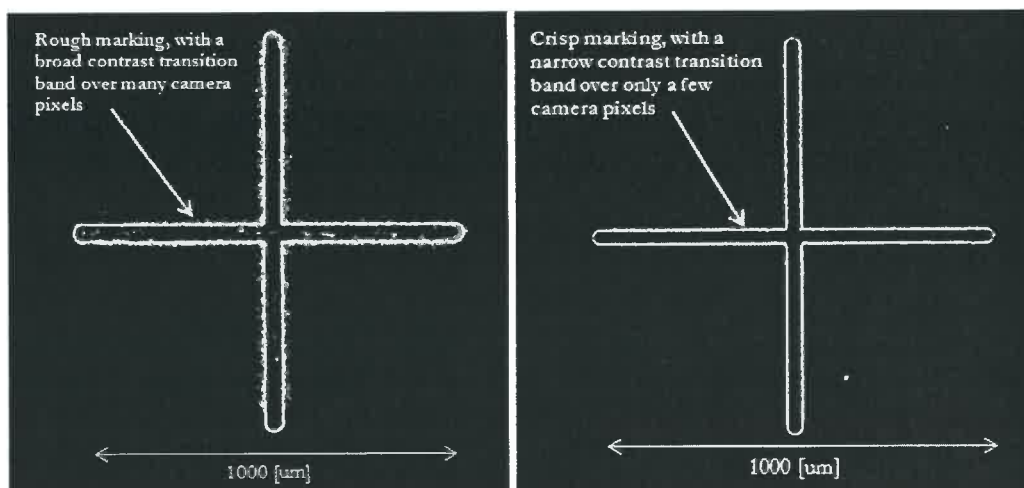


FIGURE 3. Examples of a poor quality mark (left) and a high quality mark (right) for potential use in the calibration of a scanner unit. The ~1 [mm] long crosshair marks were made using Aerotech Inc.'s AGV-14HP-1064-100T configured scanner. The marks were made on a metallic thin film coated glass substrate with a YAG fiber laser operating in pulsed mode. The high quality mark's inner width is ~40 [μm], nearly the size of the theoretical focused $1/e^2$ Gaussian laser spot diameter.

CALIBRATION RESULTS & CONCLUSIONS

Below are the documented results of the fore mentioned case study on mark-and-measure calibration. The purpose of the study was to determine and document the readily achievable marking accuracy of an Aerotech AGV-14HP-1064-100T scanner system when calibrated using Aerotech's High Precision Inspection Station via the mark-and-measure technique.

The study was not meant to define the absolute limit for marking accuracy in scanner based systems, nor was it directly a representation of the rotational accuracy of scanners within Aerotech's AGV-14HP model scan heads. The experiment was meant only to act as an example of the output accuracy that a similarly fitted laser processing system might be able to attain when attention is given to the principles and considerations discussed above.

Results

After mark-and-measure calibration, the AGV-14HP-1064-100T scanner unit achieved 2D marking accuracies on the order of singular micrometers of error. The total calibration took two calibration iterations (three marked grids) to return no improvement in accuracy. Table 1 shows a summary of the measurements taken from the final grid marked by the scanner. It is important to note that the achieved accuracy is roughly equal to the approximate measurement uncertainty of the grid. Thus, the achieved accuracy has reached the effective calibration resolution for this particular system.

Conclusions

In this case study, an AGV-14HP-1064-100T galvanometer scanner was calibrated using a

mark-and-measure methodology to give insight into the levels of marking accuracy achievable with this type of calibration approach. It was learned that the effectiveness of a mark-and-measure calibration procedure is largely dependent on the combination of substrate and laser parameters used for marking. The combination of these two factors determines the overall mark quality of the fiducials made in terms of measurability and measurement repeatability. The mark quality, in this study, was the majority contributor to measurement uncertainty, and therefore was the limiting factor in the calibration outcome.

For marking accuracy to be limited by the repeatability of the galvanometer motors themselves along with the impacts of one's ability to control the spatial relations of the marking station, measurement uncertainty must be minimized. Regardless, the achievable accuracy of the laser marking system will be the combination of the galvanometer motor's repeatability, the effects of spatial non-repeatabilities in the system setup, and the uncertainty in the measurement process. Therefore, this summation should be at or lower than the accuracy goals of the process at hand, in this instance DMLS.

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Note: This paper is a direct modification of reference [1] with practically no addition of new material. It is merely a resubmission of [1] with a reduction of its size and scope.

TABLE 1. Measurement summary of the final grid made during the Mark-and-Measure calibration of an AGV-14HP-1064-100T scanner unit.

Summary of Achieved 2D Marking Accuracy		Scanner: AGV-14HP-1064-100T
Metallic Coated Glass Substrates		
Calibrated Field Size = 40 x 40 [mm]		
Camera Resolution = 0.86 [μm/pixel]		
Measurement Uncertainty [μm pk-pk]:		2.50
X-Axis	X Marking Accuracy [μm pk-pk]:	2.65
	X Avg. Linearity [%]:	99.997
Y-Axis	Y Marking Accuracy [μm pk-pk]:	2.33
	Y Avg. Linearity [%]:	99.998
X-Y Orthogonality [arc-sec]:		0.01
Vector Sum Error [μm pk-pk]:		1.40

OPTIMAL ZERO PHASE ERROR TRACKING FEEDFORWARD CONTROL FOR AN ULTRA-PRECISION DUAL-STAGE ACTUATED WAFER STAGE

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INSTRUCTIONS

Zero phase error tracking feedforward controller (ZPETFC) has a clear advantage to improve the tracking performance of non-minimum phase systems. The traditional ZPETFC ensures the magnitude-frequency response is unity at only zero frequency, which is undesired for high precision motion systems. Therefore, an optimal ZPETFC with a concise gain compensation filter is proposed to improve the magnitude-frequency characteristics and thus the tracking performance.

In general, ZPETFC is used in 2-DOF tracking control system of close-loop inverse structure (CLIS)^[1] in most literatures, leading to higher order and more complex implementation than the plant inverse structure (PIS). However, because ZPETFC isn't the exact model-inverse of the controlled plant, PIS suffers from reduced settle performance due to excitation of the closed-loop dynamics^[2]. 2-DOF tracking control system of modified plant inverse structure (MPIS) provides a means to deal with the trade-off between the lower order of ZPETFC and shorter settling time.

Comparative simulations are conducted on the model of our developed wafer stage which employs a dual-stroke principle. The results show that the proposed optimal ZPETFC in MPIS can significantly improve the moving average (MA) of the tracking error and the standard deviation (MSD) of the tracking error, and the settling time is reduced from 11.6ms to 0ms.

OPTIMAL ZPETFC IN MPIS

In order to simultaneously achieve the low order of ZPETFC and shorter settling time, 2-DOF tracking control system of MPIS shown in figure 1 will be adopted, where $G(z^{-1})$ represents the plant, $F(z^{-1})$ represents ZPETFC aiming at approximating the model-inverse of $G(z^{-1})$, and

$C(z^{-1})$ represents the feedback controller. Given the reference command $r(t)$ and the output $y(t)$ of the plant $G(z^{-1})$, the tracking error $e(t) = r(t) - y(t)$. Compared with PIS, a prefilter $M(z^{-1})$ is added in MPIS.

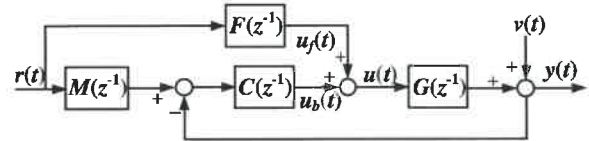


FIGURE 1. 2-DOF tracking control system of MPIS.

The plant $G(z^{-1})$ is denoted as

$$G(z^{-1}) = \frac{z^{-d} B(z^{-1})}{A(z^{-1})} = \frac{z^{-d} B_u(z^{-1}) B_s(z^{-1})}{A(z^{-1})} \quad (1)$$

where $B_u(z^{-1})$ is polynomial with unstable zeros, $B_s(z^{-1})$ is with stable zeros and d is time delay.

The optimal ZPETFC $F(z^{-1})$ adds a gain compensation filter $F_g(z^{-1}, z)$ ahead of the traditional ZPETFC and is designed as^[3]

$$F(z^{-1}) = F_g(z^{-1}, z) \cdot \frac{B_u(z)}{B_u(1)^2} \cdot \frac{z^d A(z^{-1})}{B_s(z^{-1})} \quad (2)$$

with $F_g(z^{-1}, z)$ designed as

$$F_g(z^{-1}, z) = \sum_{i=0}^N (\alpha_i (z^{-i} + z^i)) \quad (3)$$

where N is the order and $\alpha = [\alpha_0 \ \alpha_1 \ \dots \ \alpha_N]^T$ is the parameter vector of $F_g(z^{-1}, z)$. Due to the same coefficient with respect to z^i and z^{-i} in (3), $F_g(z^{-1}, z)$ is a zero-phase filter.

Let

$$\begin{aligned} R(z^{-1}) &= F(z^{-1}) G(z^{-1}) = F_g(z^{-1}, z) \frac{B_u(z) B_s(z^{-1})}{B_u(1)^2} \\ &= \left[\sum_{i=0}^N (\alpha_i (z^{-i} + z^i)) \right] \cdot \left[\sum_{j=0}^p \gamma_j (z^{-j} + z^j) \right] \end{aligned} \quad (4)$$

where γ_j is the coefficient of the polynomial $B_u(z)$ $B_u(z^{-1})/B_u(1)^2$ corresponding to z^j and z^{-j} , and P is the order.

If $M(z^{-1})=1$, i.e. PIS, the transfer function $H(z^{-1})$ from the tracking error $e(t)$ to the input $r(t)$ is described as

$$H(z^{-1}) = \frac{1 - R(z^{-1})}{1 + G(z^{-1})C(z^{-1})} \quad (5)$$

Because $F(z^{-1})$ isn't the exact model-inverse of $G(z^{-1})$, i.e. $R(z^{-1}) \neq 1$, PIS suffers from reduced settling performance due to excitation of the closed-loop dynamics. However, settling time is tightly associated with the throughput of wafer stage and thus an important performance index. MPIS is aimed at reducing settling time.

The transfer function $T(z^{-1})$ from $e(t)$ to $r(t)$ in MPIS is described as

$$T(z^{-1}) = \frac{(1 - R(z^{-1})) + (1 - M(z^{-1}))G(z^{-1})C(z^{-1})}{1 + G(z^{-1})C(z^{-1})} \quad (6)$$

Selecting $M(z^{-1}) = R(z^{-1})$, $T(z^{-1})$ is rewritten as

$$T(z^{-1}) = 1 - R(z^{-1}) \quad (7)$$

Thus, the closed-loop dynamics will not be excited in MPIS even though $R(z^{-1}) \neq 1$.

If $R(z^{-1})=1$ in MPIS, the tracking error will be completely eliminated. Here the parameter vector α of $F_g(z^{-1}, z)$ is obtained by minimizing the objective function $J(\alpha)$ as follows.

$$\min J(\alpha) = \frac{1}{2\pi} \int_{\theta_1}^{\theta_2} |R(e^{-j\theta}) - 1|^2 d\theta \quad (8)$$

where $\theta = \omega \cdot T_s$, and T_s is the sample interval. The constraint that ensures the magnitude -frequency response of $R(z^{-1})$ is unity at zero frequency can be denoted as

$$R(e^{-j\theta})|_{\theta=0} = 1 \quad (9)$$

Furthermore, a general weighting function $A(\theta)$ can be introduced in Eq.(8) to emphasize some frequency range.

The constrained L_2 -norm optimal problem described in Eq. (8) and Eq. (9) can be solved via the Lagrange method and the optimal parameter vector α of $F_g(z^{-1}, z)$ is obtained as

$$\alpha = \frac{1}{2} (\Gamma^T A \Gamma)^{-1} (2\Gamma^T \Omega - \lambda \beta) \quad (10)$$

where the definitions and the detailed derivations are elaborated at the appendix.

The above method of obtaining the optimal parameter vector α is similar to that proposed by Syh-Shiuh, Y. and H. Pau-Lo^[3], but the present

calculation is considerably simplified because of a clever transformation of the objective function. The integrals of the trigonometric function matrix and vector in [3] are associated with the coefficient γ_j of the polynomial $B_u(z)$ $B_u(z^{-1})/B_u(1)^2$, i.e. the parameters of the controlled plant, while the integrals in this paper are not. Thus, the computation of the present method is considerably decreased in the case of dealing with different plants or combining the optimal ZPETFC with on-line identification to achieve adaptive feedforward control, because the integrals can be solved in advance.

The prefilter $M(z^{-1})$ is FIR filter(see Eq.(4)). Thus the introduction of $M(z^{-1})$ doesn't affect the original close-loop stability. As the closed-loop dynamics will not be excited in MPIS even though $R(z^{-1}) \neq 1$, the proposed optimal ZPETFC in MPIS realizes the same settling time performance as in CLIS and the same order as in PIS. However, errors in the acceleration/deceleration phase in MPIS are larger than those in PIS because of the absent filtering by $1/(1 + G(z^{-1})C(z^{-1}))$.

SIMULATION RESULTS

In this section, the proposed algorithm is assessed with comparative simulations on the identification model of our developed wafer stage shown in figure 2. The developed wafer stage employs a dual-stroke principle with a short-stroke stage (SS) for accurate positioning (nanometer scale) and a long-stroke stage (LS) for coarse positioning (micron scale). SS is driven by eight voice coil motors to achieve 6-DOF motion and the magnetic gravity compensator is to balance the gravity of SS. LS is a planar actuator suspended above the surface of a Halbach permanent magnet array by air bearings.

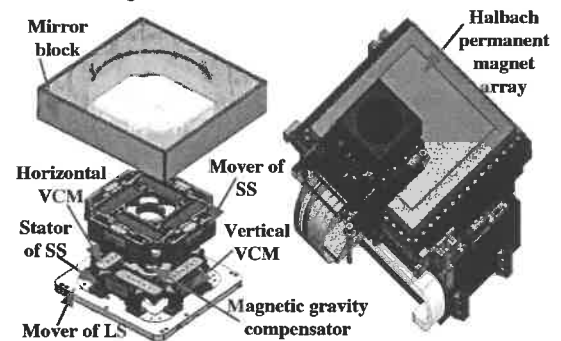


FIGURE 2. Schematic of the developed wafer stage.

The measured frequency response of SS in the scanning direction shown in figure 3 reveals that it is characterized by double integrator behavior in the low-frequency range along with higher order dynamics. The measured unstable zeros of SS in the scanning direction are $-1.7407 \pm 2.3091i$, $0.6045 \pm 1.2476i$, in the left-half plane (LHP) and the right-half plane (RHP) respectively.

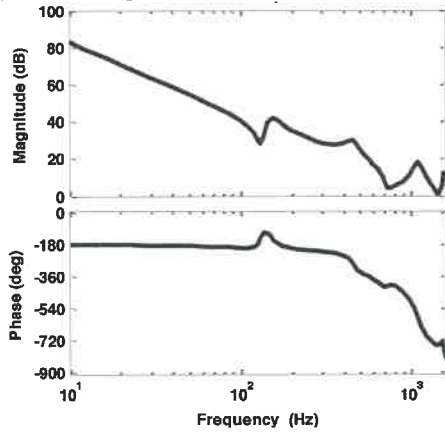


FIGURE 3. Measured frequency response of SS in the scanning direction.

The frequency responses of $R(z^{-1})$ obtained by the traditional ZPETFC and the optimal ZPETFC (selecting $N=10$, $\theta_1=0$, $\theta_2=\pi$ and $N=20$, $\theta_1=0$, $\theta_2=\pi$) are shown in figure 4. Due to existence of both LHP zeros and RHP zeros, magnitude of $R(z^{-1})$ by the traditional ZPETFC firstly decreases to -16.8dB at 910Hz and then increases gradually. However, magnitude of $R(z^{-1})$ obtained by the optimal ZPETFC is within ± 0.7 dB if $N=10$ and within ± 0.02 dB if $N=20$. It is obvious that the larger N , the better magnitude-frequency characteristics of $R(z^{-1})$, but the more complex implementation.

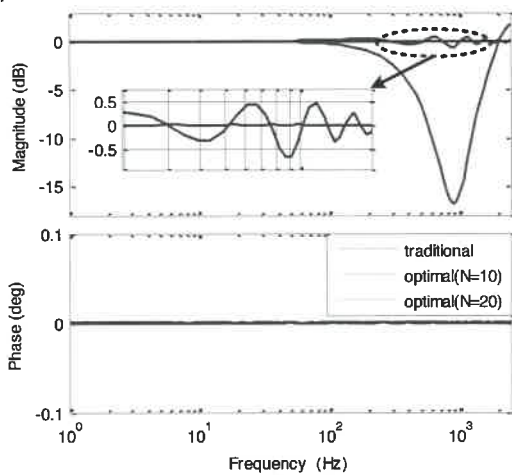


FIGURE 4. Frequency responses of $R(z^{-1})$ obtained by the traditional and optimal ZPETFC.

The fourth order reference profile with a cruising speed of 0.25m/s and a maximum acceleration/deceleration of 10 m/s² shown in figure 5 is adopted. The sample interval T_s is 200 μ s.

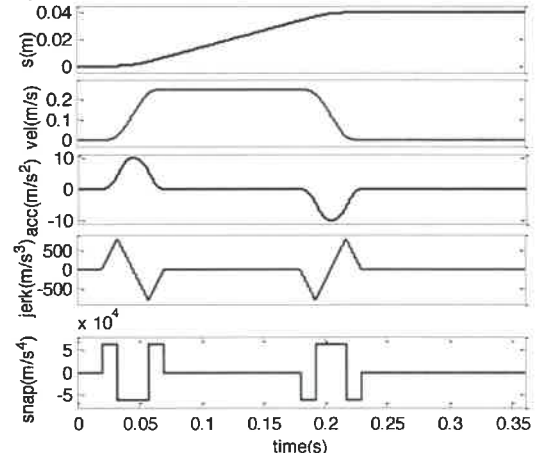


FIGURE 5. Fourth order reference profile.

In general, the main performance indexes for wafer stage are MA of the tracking error, MSD of the tracking error and settling time^[4]. Indicative of the overlay error contribution is MA representing the low-frequency part of the tracking error, while a measure for contrast loss in the imaging process is MSD representing the high-frequency part of the tracking error. Settling time represents the time period for the tracking error converging to a specified accuracy after acceleration phase, which is tightly linked with the throughput of wafer stage. MA and MSD are described as follows.

$$MA(k) = \frac{1}{n} \sum_{i=k-n/2}^{k+n/2-1} error(k) \quad (11)$$

$$MSD(k) = \sqrt{\frac{1}{n} \sum_{i=k-n/2}^{k+n/2-1} (error(k) - MA(k))^2}$$

The comparison of the tracking error between the traditional ZPETFC in PIS and the optimal ZPETFC ($N=20$) in PIS is shown in Figure 6. The simulation results demonstrate that the proposed optimal ZPETFC considerably improves the tracking performance and decreases settling time. The larger N , the better performance (not elaborated here). However, due to excitation of the closed-loop dynamics in PIS, the settling time is larger than the desired value 10ms even when $N=20$ and larger N is undesired.

Figure 7 presents the comparison of the tracking error between the optimal ZPETFC ($N=20$) in PIS and the optimal ZPETFC ($N=20$) in MPIS. The proposed optimal ZPETFC in MPIS realizes

smaller settling time and the comparative level of tracking performance in constant speed phase compared with the proposed optimal ZPETFC in PIS. However, the tracking error in the acceleration/deceleration phase in MPIS are larger than those in PIS (but less than the traditional ZPETFC in PIS or MPIS, not elaborated here) because of the absent filtering by $1/(1+G(z^{-1})C(z^{-1}))$. Note that the illumination starts during the constant speed phase. This is the region where the tracking performance of wafer stage should be achieved. Therefore, the increased tracking errors in the acceleration/deceleration phase in MPIS will not notably affect the performance of the wafer stage. Table 1 presents the detailed tracking performance indexes. The settling time is calculated on the criterion that MA is less than 3nm and MSD is less than 6nm.

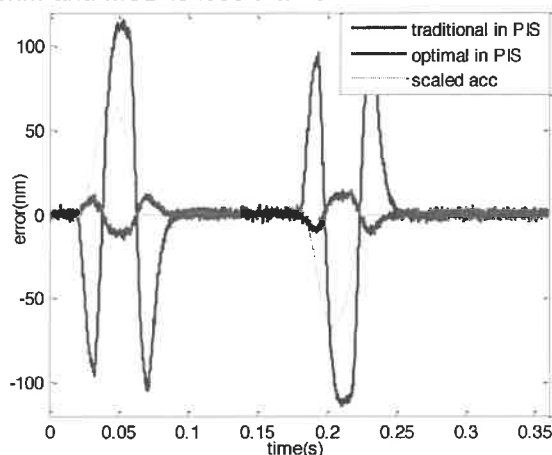


FIGURE 6. The Comparison of the tracking error between the traditional ZPETFC in PIS and the optimal ZPETFC ($N=20$) in PIS.

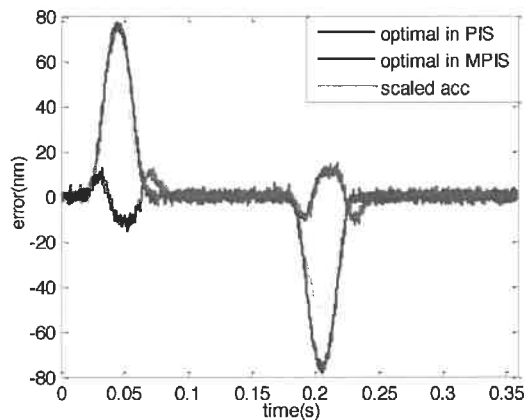


FIGURE 7. The Comparison of the tracking error between the optimal ZPETFC ($N=20$) in PIS and the optimal ZPETFC ($N=20$) in MPIS.

TABLE 1. The detailed tracking performance indexes.

Performance indicators	Traditional in PIS	Optimal in PIS	Optimal in MPIS	Units
Settling time	21.0	11.6	0	[ms]
$\ error\ _{\infty}$	115.7	14.5	77.1	[nm]
$\ MA\ _{\infty}$	110.8	11.4	73.4	[nm]
$\ MSD\ _{\infty}$	61.4	6.1	14.7	[nm]

Conclusion

An optimal ZPETFC for a non-minimum phase ultra-precision dual-stage actuated wafer stage has been studied in this paper. By applying the Lagrange method to solve a constrained L_2 -norm optimal problem a concise gain compensation filter of the proposed optimal ZPETC is obtained. Due to a clever transformation of the objective function, the calculation of the parameter vector of the gain compensation filter is greatly simplified, leading to dramatic decrease of computation especially in the case of dealing with different plants or combining the optimal ZPETFC with on-line identification to achieve adaptive feedforward control. The proposed optimal ZPETC maintains the characteristic of zero-phase and significantly enhances the tracking performance of the tracking control system (see table 1 for detailed info.). Moreover, MPIS provides a means to deal with the trade-off between the low order of ZPETFC and non-excitation of the closed-loop dynamics. The proposed optimal ZPETFC in MPIS realizes smaller settling time (reducing from 11.6ms to 0ms) and the comparative level of tracking performance in constant speed phase compared with that in PIS, which are desired for wafer stage.

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Appendix

We adopt the Lagrange method to solve the constrained L_2 -norm optimal problem described in Eq. (8) and Eq. (9) and the Lagrange function $L(\alpha, \lambda)$ is defined as

$$L(\alpha, \lambda) = \frac{1}{2\pi} \int_{\theta_1}^{\theta_2} |R(e^{-j\theta}) - 1|^2 d\theta + \lambda (R(e^{-j\theta})|_{\theta=0} - 1) \quad (A-1)$$

where λ is the Lagrange factor. According to the following equation

$$e^{j\theta} + e^{-j\theta} = 2 \cos \theta \quad (A-2)$$

$R(e^{j\theta})$ is rewritten as

$$\begin{aligned} R(e^{j\theta}) &= \left[\sum_{i=0}^N (\alpha_i (z^{-i} + z^i)) \right] \cdot \left[\sum_{j=0}^P \gamma_j (z^{-j} + z^j) \right] \\ &= \begin{pmatrix} 2 \cos(\theta) & 2 \cos(2\theta) & 2 \cos(3\theta) & \cdots & 2 \cos((N+P)\theta) \end{pmatrix} \cdot \\ &\quad \begin{pmatrix} 2\gamma_0 & \gamma_1 & \gamma_2 & \gamma_3 & \cdots & \gamma_N \\ 2\gamma_1 & 2\gamma_0 + \gamma_2 & \gamma_1 + \gamma_3 & \gamma_2 + \gamma_4 & \cdots & \gamma_{N-1} + \gamma_{N+1} \\ 2\gamma_2 & \gamma_1 + \gamma_3 & 2\gamma_0 + \gamma_4 & \gamma_1 + \gamma_5 & \cdots & \gamma_{N-2} + \gamma_{N+2} \\ 2\gamma_3 & \gamma_2 + \gamma_4 & \gamma_1 + \gamma_5 & 2\gamma_0 + \gamma_6 & \cdots & \gamma_{N-3} + \gamma_{N+3} \\ \vdots & \vdots & \vdots & \vdots & \ddots & \vdots \\ 2\gamma_N & \gamma_{N-1} + \gamma_{N+1} & \gamma_{N-2} + \gamma_{N+2} & \gamma_{N-3} + \gamma_{N+3} & \cdots & \gamma_0 + \gamma_{2N} \\ \vdots & \vdots & \vdots & \vdots & \ddots & \vdots \\ 2\gamma_{N+P} & \gamma_{N+P-1} + \gamma_{N+P+1} & \gamma_{N+P-2} + \gamma_{N+P+2} & \gamma_{N+P-3} + \gamma_{N+P+3} & \cdots & \gamma_P + \gamma_{2N+P} \end{pmatrix} \cdot \begin{pmatrix} \alpha_0 \\ \alpha_1 \\ \alpha_2 \\ \alpha_3 \\ \vdots \\ \alpha_N \end{pmatrix} \\ &= \Theta^T \Gamma \alpha \end{aligned} \quad (A-3)$$

With $\gamma_{P+1}, \gamma_{P+2}, \dots, \gamma_{N+P}=0$ and

$$\Theta = \begin{pmatrix} 2 & 2\cos(\theta) & 2\cos(2\theta) & 2\cos(3\theta) & \dots & 2\cos((N+P)\theta) \end{pmatrix}_{[(N+P+1) \times 1]}^T$$

$$\Gamma = \begin{pmatrix} 2\gamma_0 & \gamma_1 & \gamma_2 & \gamma_3 & \dots & \gamma_N \\ 2\gamma_1 & 2\gamma_0 + \gamma_2 & \gamma_1 + \gamma_3 & \gamma_2 + \gamma_4 & \dots & \gamma_{N-1} + \gamma_{N+1} \\ 2\gamma_2 & \gamma_1 + \gamma_3 & 2\gamma_0 + \gamma_4 & \gamma_1 + \gamma_5 & \dots & \gamma_{N-2} + \gamma_{N+2} \\ 2\gamma_3 & \gamma_2 + \gamma_4 & \gamma_1 + \gamma_5 & 2\gamma_0 + \gamma_6 & \dots & \gamma_{N-3} + \gamma_{N+3} \\ \vdots & \vdots & \vdots & \vdots & \ddots & \vdots \\ 2\gamma_N & \gamma_{N-1} + \gamma_{N+1} & \gamma_{N-2} + \gamma_{N+2} & \gamma_{N-3} + \gamma_{N+3} & \dots & \gamma_0 + \gamma_{2N} \\ \vdots & \vdots & \vdots & \vdots & \ddots & \vdots \\ 2\gamma_{N+P} & \gamma_{N+P-1} + \gamma_{N+P+1} & \gamma_{N+P-2} + \gamma_{N+P+2} & \gamma_{N+P-3} + \gamma_{N+P+3} & \dots & \gamma_P + \gamma_{2N+P} \end{pmatrix}_{[(N+P+1) \times (N+1)]}$$

$$\alpha = (\alpha_0 \quad \alpha_1 \quad \alpha_2 \quad \alpha_3 \quad \dots \quad \alpha_N)_{[(N+1) \times 1]}^T$$

Therefore,

$$\left| R(e^{-j\theta}) - 1 \right|^2 = R^2(e^{j\theta}) - 2R(e^{j\theta}) + 1 = \alpha^T \Gamma^T \Theta \Theta^T \Gamma \alpha - 2\Theta^T \Gamma \alpha + 1 \quad (\text{A-5})$$

and

$$R(e^{-j\theta}) \Big|_{\theta=0} = \begin{pmatrix} 2 & 2 & 2 & \dots & 2 \end{pmatrix} \begin{pmatrix} \alpha_0 \\ \alpha_1 \\ \alpha_2 \\ \vdots \\ \alpha_N \end{pmatrix} = \beta^T \alpha \quad (\text{A-6})$$

with $\beta = [2 \ 2 \ 2 \ \dots \ 2]^T$ and the dimension of β is $(N+1)$.

Substituting (A-5) and (A-6) into (A-1), we could obtain

$$L = \alpha^T \Gamma^T A \Gamma \alpha - 2\Omega^T \Gamma \alpha + \frac{1}{2\pi} (\theta_2 - \theta_1) + \lambda (\beta^T \alpha - 1) \quad (\text{A-7})$$

Where

$$A = \frac{1}{2\pi} \int_{\theta_1}^{\theta_2} \Theta \Theta^T d\theta$$

$$\Omega = \frac{1}{2\pi} \int_{\theta_1}^{\theta_2} \Theta d\theta \quad (\text{A-8})$$

Let

$$\begin{cases} \frac{\partial L}{\partial \alpha} = 2\Gamma^T A \Gamma \alpha - 2\Gamma^T \Omega + \lambda \beta = 0 \\ \frac{\partial L}{\partial \lambda} = \beta^T \alpha - 1 = 0 \end{cases} \quad (\text{A-9})$$

Via solving the equation set (A-9), the Lagrange factor λ is obtained as

$$\lambda = 2 \frac{\beta^T (\Gamma^T A \Gamma)^{-1} \Gamma^T \Omega - 1}{\beta^T (\Gamma^T A \Gamma)^{-1} \beta} \quad (\text{A-10})$$

Finally, Substituting (A-10) into the first equation of (A-9), the optimal parameter vector α of $F_g(z^{-1}, z)$ is obtained as

$$\alpha = \frac{1}{2} (\Gamma^T A \Gamma)^{-1} (2\Gamma^T \Omega - \lambda \beta) \quad (\text{A-11})$$

UNCERTAINTY ESTIMATION OF A FIVE-AXIS MACHINE TOOL CALIBRATION USING THE ADAPTIVE MONTE CARLO METHOD

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INTRODUCTION

Calibration of a five-axis machine tool allows verifying its ability to machine with the required accuracy and, if deemed necessary, correcting it. Numerous methods, using different kinds of devices and artefacts, are applied to estimate the machine error parameters [1].

In order to be able to compare those methods, the results of calibration must also be evaluated. That can be obtained through different approaches. Bringmann et al. [2] used the Monte Carlo method (MCM) to predict workpiece feature errors with and without calibration with the objective of choosing the most optimal calibration method. In [9] the uncertainties on the machine geometric errors parameters of the model-based calibration method were estimated using general MCM. Other researchers also applied MCM based on The Guide to the Expression of Uncertainty in Measurement Supplement 1 (GUM S1) [7]. Andolfatto et al. [8] used the adaptive MCM to estimate uncertainty on machine tool axis location errors with confidence intervals. Santolaria et al. [3] conducted robot kinematic calibration and through simulation estimated the uncertainty on the calibration results.

All of these methods allow estimating many geometric errors simultaneously. That is why the covariance between them should also be considered when the uncertainties are calculated. The uncertainty estimation of a multi-output method is proposed in The Guide to the Expression of Uncertainty in Measurement Supplement 2 (GUM S2) [4].

In this paper the uncertainties for two calibration methods, the reconfigurable uncalibrated master balls artefact (RUMBA) method [5] and scale enriched master balls artefact (SAMBA) method [6] are estimated for different numbers of master balls. Since both of them are based on multistage and iterative calculations, the adaptive MCM is followed.

CALIBRATION METHOD

FIGURE 1 shows the SAMBA artefact used for the study. It consists of four master balls 12.7 mm in diameter mounted on rods and a scale bar with a length of 304.6686 mm (which is not used for a RUMBA).

Calibrating the machine tool using the RUMBA or SAMBA methods requires probing a number of master balls (for SAMBA, also probing a scale bar at least once) at different machine rotary axis indexations (input data).

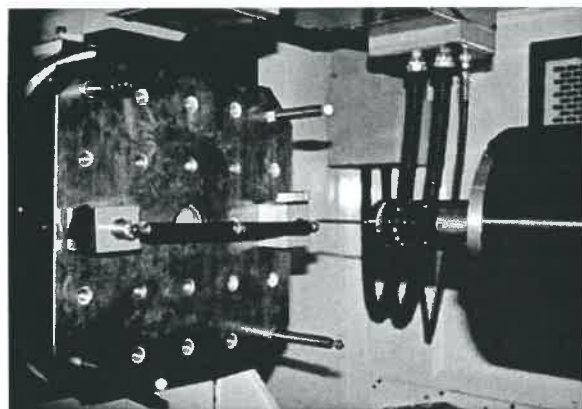


FIGURE 1. SAMBA artefact being probed using the Renishaw probe MP700 on a Mitsui Seiki HU40-T machine tool.

For each indexation, each of the four balls is probed in five points, which allows calculating the X-, Y- and Z-axis readings corresponding to positioning the stylus tip at the ball center and comparing it with its nominal position. The difference between those two sets of coordinates provides the initial model prediction error.

The identification of the geometric error parameters requires building the kinematic error model of the machine. That allows calculating the predicted stylus tip position relative to the probed ball center position.

The Newton-Gauss approach is used to estimate the required changes to the machine geometric error parameters in order to reduce the error between the prediction of the model and the observation and so better explain the difference with the prediction of those positions.

MEASUREMENTS AND SIMULATION

The investigated five-axis machine is the Mitsui Seiki HU40-T. Its topology is depicted in *FIGURE 2*. The analyzed machine is modeled using thirteen joint link errors with values estimated during previous calibration [5]. These machine geometric errors parameters, listed in *TABLE 1*, constitute the output data.

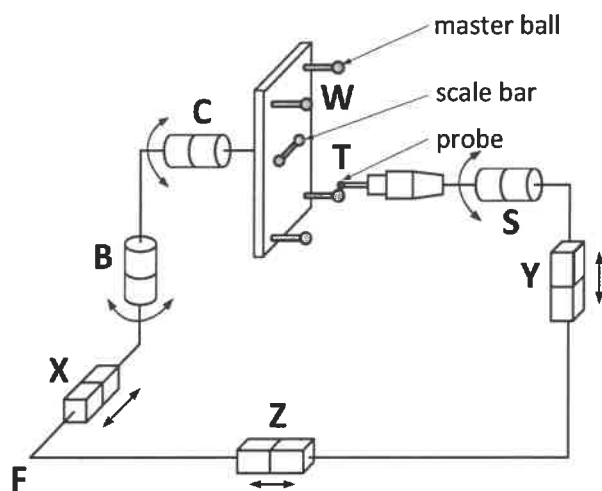


FIGURE 2. Five-axis CNC machine model with the topology WCBXFZYST; W – workpiece, T – tool, F – machine foundation, B, C – rotary axes around the Y and Z axes respectively, X, Y, Z – machine linear axes and S – spindle.

The probing is simulated for seven different rotary axis indexations ($[B, C] = [90, 270], [60, 180], [30, 90], [0, 0], [-90, -270], [-60, -180],$ and $[-30, -90]$ deg) using $n=1,2,3,4$ balls for both SAMBA and RUMBA. When the former is applied the scale bar is measured once for $B=C=0$ deg.

In order to estimate the distribution and associated uncertainty of the input data repeated SAMBA measurements are performed on the analyzed machine. That allows calculating a covariance matrix of the input data, its probability density functions (PDFs) and the joint PDF.

TABLE 1. Identified geometric errors parameters

Symbol	Description
E_{AOB}	out-of-squareness of the B-axis relative to the Z-axis (μrad)
E_{COB}	out-of-squareness of the B-axis relative to the X-axis (μrad)
E_{XOC}	distance between the B and C axes (μm)
E_{AOC}	out-of-squareness of the C-axis relative to the B-axis (μrad)
E_{BOC}	out-of-squareness of the C-axis relative to the X-axis (μrad)
E_{BOZ}	out-of-squareness of the Z-axis relative to the X-axis (μrad)
E_{AOY}	out-of-squareness of the Y-axis relative to the Z-axis (μrad)
E_{COY}	out-of-squareness of the Y-axis relative to the X-axis (μrad)
E_{XOS}	X offset of the spindle relative to the B-axis (μm)
E_{YOS}	Y offset to the spindle relative to the C-axis (μm)
E_{XX}	positioning linear error term of the X-axis ($\mu\text{m}/\text{m}$)
E_{YY}	positioning linear error term of the Y-axis ($\mu\text{m}/\text{m}$)
E_{ZZ}	positioning linear error term of the Z-axis ($\mu\text{m}/\text{m}$)

UNCERTAINTY ESTIMATION

The MCM described in [7] allows estimating uncertainties without defining the partial derivatives of the measurement model. In order to calculate the uncertainty using MCM M arrays of the N input data are created by sampling their joint PDF. For each of the M input arrays the m output quantities are calculated. M vectors of the output quantity are used to calculate the output value vector by calculating its average. Associated covariance matrix U_y (with squares of the output quantity uncertainties on its diagonal), coverage factor k_p for hype-ellipsoid and k_q hyper-rectangular coverage regions for the coverage probability p are also calculated.

Although the number of MCM trials can be set before conducting calculations, in this paper the adaptive MCM is performed, thus the final number of MC trials depends on the results of a stability assessment.

The first step of the adaptive MCM is performing $h=10$ times $M=10^4$ MC trials. For each $r=1, \dots, h$

subsets of the output quantity estimate $y(r)$ and its standard deviation $s_y^{(r)}$, maximum eigenvalue $\lambda_{max}^{(r)}$ of the correlation matrix $R_y^{(r)}$, $k_p^{(r)}$ and $k_q^{(r)}$ are calculated. The $h=10$ estimates subsets of the output quantities are used to calculate their standard deviations (for each of the m output quantity) as one of the stability parameters. In a similar way, the remaining stability parameters standard deviations (of results variances s_{uyj} , maximum eigenvalue $s_{\lambda_{max}}$ and coverage factors s_{kp} and s_{kq}) are estimated. The obtained values are multiplied by two and compared with their corresponding required numerical tolerances. If at least one of the doubled standard deviations is greater than its numerical tolerance, the value of h is increased by 1 and a new set of M output data is calculated. The procedure is repeated until the results reach the demanded stability.

Then all the hM results are used to estimate the output quantities averages, the associated covariance matrix, the coverage factors and finally the uncertainties.

RESULTS

The parameters and their uncertainties have been calculated for 1,2,3 and 4 master balls. The identification results and their expanded uncertainties for $p=0.95$ are listed in TABLE 2. The coverage factor k_q varies between 2.9 and 3.0. The average time of performing adaptive MCM is around 20 hours and depends on the number of master balls used. Both, RUMBA and SAMBA, give similar estimated results. In all the presented adaptive MCM cases the results reached the required stability after $hM=10^5$ MC trials.

TABLE 2. Five-axis CNC machine tool calibration results and their uncertainties for RUMBA and SAMBA (* MCM with $M=10^4$ trials).

Parameter	Calibr. result	Expanded uncertainty ($p=0.95$)							
		RUMBA				SAMBA			
		$n=1^*$	$n=2^*$	$n=3$	$n=4$	$n=1$	$n=2$	$n=3$	$n=4$
E_{AOB} (μrad)	0.9	± 1.1	± 2.7	± 4.4	± 3.4	± 1.0	± 3.4	± 3.2	± 3.3
E_{COB} (μrad)	-1.5	± 3.6	± 3.7	± 3.2	± 2.7	± 3.6	± 2.9	± 3	± 2.5
E_{XOC} (μm)	-102.2	± 1.0	± 0.90	± 1.0	± 1.0	± 0.90	± 0.90	± 1.2	± 1.0
E_{AOC} (μrad)	3.9	± 7.2	± 7.5	± 5.0	± 3.1	± 7.1	± 4.0	± 3.6	± 2.9
E_{BOC} (μrad)	19.9	± 8.0	± 3.2	± 4.6	± 4.8	± 8.0	± 3.6	± 6.3	± 4.5
E_{BOZ} (μrad)	-37.5	± 16	± 8.2	± 4.0	± 4.0	± 16	± 4.7	± 3.4	± 3.9
E_{AOY} (μrad)	-8.8	± 6.5	± 6.2	± 9.6	± 9.6	± 6.5	± 6.0	± 13	± 9.0
E_{COY} (μrad)	23.9	± 12	± 7.3	± 3.1	± 2.9	± 12	± 4.8	± 4.1	± 2.8
E_{XOS} (μm)	-97.1	± 5.3	± 7.3	± 8	± 7.5	± 5.1	± 7.0	± 8.7	± 7.0
E_{YOS} (μm)	15.7	± 2.6	± 2.6	± 2.3	± 3.0	± 2.5	± 3.9	± 2.3	± 2.8
E_{XX} ($\mu\text{m}/\text{m}$)	-45.2	-	-	-	-	± 28	± 29	± 29	± 29
$E_{YY} - E_{XX}$ or E_{YY} ($\mu\text{m}/\text{m}$)	50.5 5.3	± 9.6 -	± 4.6 -	± 4.4 -	± 2.2 -	- ± 34	- ± 38	- ± 27	- ± 29
$E_{ZZ} - E_{XX}$ or E_{ZZ} ($\mu\text{m}/\text{m}$)	25.0 -20.5	± 6.8 -	± 8.3 -	± 12 -	± 12 -	- ± 32	- ± 34	- ± 35	- ± 38
k_q		2.9	3.0	2.9	3.0	2.9	2.9	2.9	2.9

The uncertainties calculated when one master ball is used do not differ a lot between the SAMBA and RUMBA results. For some of the analyzed parameters the uncertainties are lower when more master balls are used. The biggest decrease of the uncertainty, when the second master ball is added, is observed for E_{BOC} , E_{BOZ} and E_{COY} (60%, 50% and 40% for RUMBA and 55%, 59% and 68% for SAMBA). Lowering uncertainty due to the higher number of master balls used is also seen for E_{AOC} , E_{COB} and for the difference $E_{YY} - E_{XX}$. The opposite situation can

be observed for E_{AOB} , E_{XOS} (with the maximum value reached for $n=3$). For E_{AOY} the uncertainty decreases after adding the second master ball and increases when the third and fourth one.

For some of the parameters the uncertainties of results obtained with SAMBA are smaller when compared with RUMBA (E_{COB} , E_{AOC} , E_{COY}). The uncertainty of the E_{XOC} almost does not differ between all four configurations and two calibration methods used. The X-axis scale error E_{XX} (which cannot be estimated for RUMBA due

to the lack of scale bar) does not differ significantly for different number of master balls included in the SAMBA artefact. This is consistent with the calculation method of this error, which is a function of the scale bar probing and its true length, thus it is not influenced by the master balls probing.

The lowest relative uncertainty is observed for E_{XOC} and is not higher than 1%. This value is also low for E_{XOS} and does not exceed 8%. For some of the errors the relative uncertainty can reach even 640% (E_{YY}), 240% (E_{COB}) and 190% (E_{AOC}).

CONCLUSION

Two calibration methods were evaluated by calculating their results uncertainties as a function of the number of master balls used through the adaptive MCM. This uncertainty estimation method, although time-consuming, proves to be very useful when an iterative measurement model is being analyzed.

Although not all the uncertainties are becoming lower when more balls are used, using 2 master balls instead of 1 gives a significant improvement of most of the estimated uncertainties for both the RUMBA and SAMBA methods.

Unexpectedly, the results show that increasing the number of master balls used (to more than 2), although providing more probing results, may cause an increase of some uncertainty values.

The presented uncertainty estimation method can be used as one of the factors when the SAMBA and RUMBA method strategies (e.g. number and configuration of SAMBA balls, number and order of the rotary axis indexations combination) are being evaluated and optimized.

ACKNOWLEDGMENTS

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SIMPLE AND ADAPTIVE DISCRETE-TIME ERROR CORRECTION FOR THE ULTRA-PRECISION S-CURVE MOTION PROFILE GENERATION

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INSTRUCTIONS

In the practical point-to-point motion control field, the S-curve motion profile is one of the simplest and most widely-used motion profiles [1], [2], [3], [4], [5]. It produces S-shaped velocity profile with limited velocity, acceleration, and jerk to achieve fast movement as well as low vibration. The typical S-curve profile is shown in Figure 1. For such profile, one of key issues of improving endpoint accuracy has attracted much attention and a lot of sound works have been developed from the dynamic perspectives such as residual vibration reduction. Deviating from the dynamic methods, our paper considers a practically inevitable static position accuracy loss which is caused by digital implementing of the position profile generation in the discrete-time domain. This discrete-time error of endpoint position (DEEP) is always ignored in most applications where the sampling period is short enough to make the calculated endpoint position error much smaller than the required accuracy. However, in ultra-precision motion applications, this error may need to be carefully considered and precisely corrected. In this paper, the DEEP is studied and a simple and adaptive algorithm to correct the DEEP is presented.

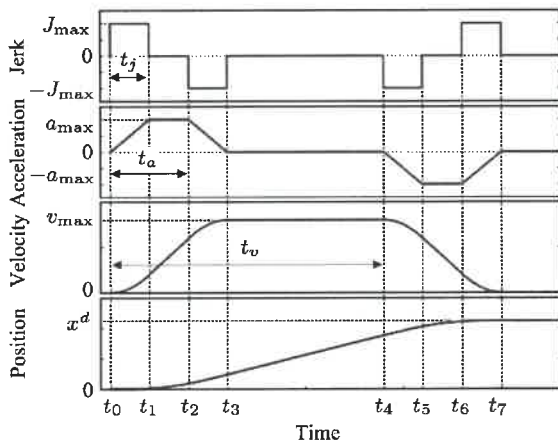


FIGURE 1. S-curve profile and its notations.

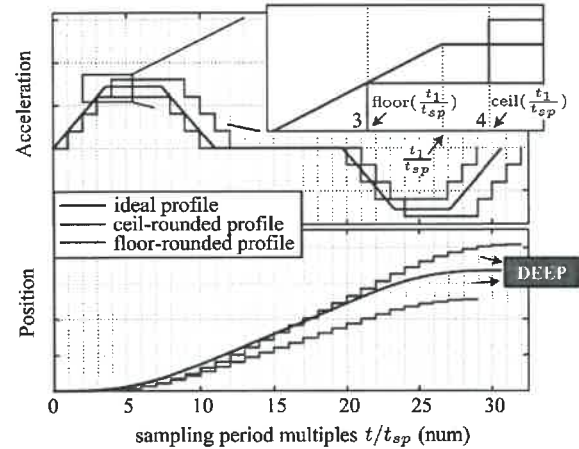


FIGURE 2. DEEP caused by time discretization.

PROBLEM FORMULATION

S-curve is realized using a two-step procedure:

S-curve profile planning in continuous-time domain: calculate switching time t_i ($i = 1, \dots, 7$) to determine the continuous profile $x^s(\cdot)$, according to the given parameters: target distance x^d , jerk limit J_{lim} , acceleration limit a_{lim} and velocity limit v_{lim} .

S-curve profile generation in discrete-time domain: round-off the switching time based on the given sampling interval t_{sp} towards the next higher or equal integer multiples of t_{sp} (so-called the ceil-round-off), or towards the last lower or equal integer multiples of t_{sp} (so-called the floor-round-off), as

$$t_i' = \text{round-off} \left(\frac{t_i}{t_{sp}} \right) \times t_{sp}, \quad (1)$$

and then calculate the discrete profile sequence $x^e(\cdot)$ in discrete-time domain.

This time discretization always causes the aforementioned DEEP. An intuitive example is shown in Figure 2. It is seen that t_1 which is about $3.5t_{sp}$ needs to be rounded-off towards $4t_{sp}$ or $3t_{sp}$, which leads to an undesired profile deformation (the position error) especially in the endpoint. This DEEP may be unacceptable in precision

motion applications. To address this problem, an additional correction step is crucial for achieving the required accuracy.

Objective: correct DEEP.

Constraints: keep velocity, acceleration, and jerk profiles within their predefined limits.

ADAPTIVE DEEP CORRECTION ALGORITHM

In this section, we develop a quite simple and adaptive DEEP correction algorithm by using a three-step procedure:

Continuous-time Profile Planning

The plan of S-curve in continuous-time domain can be referred in [1]. The unified mathematical calculations of the S-curve are determined by $t_i (i = 1, \dots, 7)$ as

$$\ddot{x}^s(t) = \begin{cases} J_{\lim}(t - t_0) & \text{for } t_0 \leq t \leq t_1 \\ a_{\lim} & \text{for } t_1 \leq t \leq t_2 \\ -J_{\lim}(t - t_3) & \text{for } t_2 \leq t \leq t_3 \\ 0 & \text{for } t_3 \leq t \leq t_4 \\ -J_{\lim}(t - t_4) & \text{for } t_4 \leq t \leq t_5 \\ -a_{\lim} & \text{for } t_5 \leq t \leq t_6 \\ J_{\lim}(t - t_7) & \text{for } t_6 \leq t \leq t_7 \end{cases} \quad (2)$$

$$\dot{x}^s(t) = \int_{t_0}^t \ddot{x}^s(\tau) d\tau, \quad x^s(t) = \int_{t_0}^t \dot{x}^s(\tau) d\tau. \quad (3)$$

Depending on the values of the parameters x^d , $J_{\lim}$, $a_{\lim}$ and $v_{\lim}$, the shape of the S-curve acceleration profile $\ddot{x}^s(\cdot)$ takes one of four cases shown in Figure 3, and the formulations of $t_i (i = 1, \dots, 7)$ can be calculated respectively. To simplify the analytic formulations, the authors [1] have introduced three critical time durations, i.e. the max jerk duration t_j , max acceleration duration t_a , and max velocity duration t_v as shown in Figure 1. And the switching times can be determined as

$$\begin{aligned} t_1 &\triangleq t_j, t_2 \triangleq t_a, t_3 \triangleq t_j + t_a, t_4 \triangleq t_v, \\ t_5 &\triangleq t_j + t_v, t_6 \triangleq t_a + t_v, t_7 \triangleq t_j + t_a + t_v, \end{aligned} \quad (4)$$

where

$$\text{Case A } \left(x^d \leq \min \left\{ \frac{2a_{\lim}^3}{J_{\lim}^2}, \frac{2v_{\lim}^{3/2}}{J_{\lim}^{1/2}} \right\} \right):$$

$$t_j \triangleq \left(\frac{x^d}{2J_{\lim}} \right)^{1/3}, t_a \triangleq t_j, t_v \triangleq 2t_j, \quad (5)$$

$$\text{Case B } \left(v_{\lim} \leq \min \left\{ \frac{a_{\lim}^2}{J_{\lim}}, \sqrt{\frac{1}{2}(x^d)^2 J_{\lim}} \right\} \right):$$

$$t_j \triangleq \left(\frac{v_{\lim}}{J_{\lim}} \right)^{1/2}, t_a \triangleq t_j, t_v \triangleq \frac{x^d}{v_{\lim}}, \quad (6)$$

$$\text{Case C } \left(\frac{2a_{\lim}^3}{J_{\lim}^2} \leq x^d \leq \frac{v_{\lim}^2}{a_{\lim}} + \frac{v_{\lim} a_{\lim}}{J_{\lim}} \right):$$

$$t_j \triangleq \frac{a_{\lim}}{J_{\lim}}, t_a \triangleq \sqrt{\frac{x^d}{a_{\lim}} + \frac{a_{\lim}^2}{4J_{\lim}^2}} - \frac{a_{\lim}}{2J_{\lim}}, t_v \triangleq t_j + t_a, \quad (7)$$

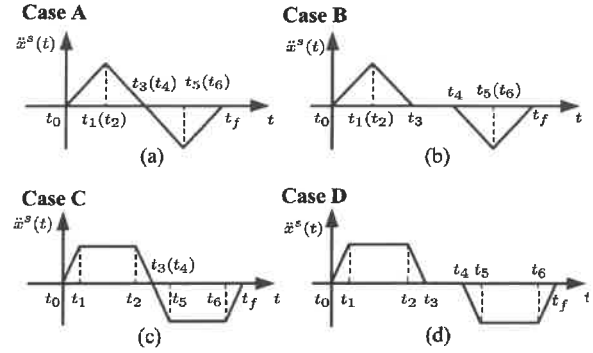


FIGURE 3. S-curve profile categories. (a) Case A. (b) Case B. (c) Case C. (d) Case D.

$$\text{Case D } \left(x^d \geq \frac{v_{\lim}^2}{a_{\lim}} + \frac{v_{\lim} a_{\lim}}{J_{\lim}} \text{ and } v_{\lim} \geq \frac{a_{\lim}^2}{J_{\lim}} \right):$$

$$t_j \triangleq \frac{a_{\lim}}{J_{\lim}}, t_a \triangleq \frac{v_{\lim}}{a_{\lim}}, t_v \triangleq \frac{x^d}{v_{\lim}}. \quad (8)$$

By using the three critical time durations t_j , t_a and t_v , the following formulations can be got:

$$J_{\max} = J_{\lim}, \quad (9)$$

$$a_{\max} = J_{\lim} \cdot t_j, \quad (10)$$

$$v_{\max} = J_{\lim} \cdot t_j \cdot t_a, \quad (11)$$

$$x^d = J_{\lim} \cdot t_j \cdot t_a \cdot t_v. \quad (12)$$

It must be noted that $J_{\max}$, $a_{\max}$, $v_{\max}$ denote the maximal jerk, maximal acceleration and maximal velocity respectively, which would be either less than or equal with the given physical limits $J_{\lim}$, $a_{\lim}$, $v_{\lim}$. For clear description, we name the $J_{\max}$, $a_{\max}$, $v_{\max}$, x^d as the profile *eigenvalues*.

Discrete-time Acceleration Profile Sequence Generating

In this paper, we round-off critical time durations t_j , t_a , t_v to t'_j , t'_a , t'_v , by using ceil-round-off (The reason of selection will be presented in the next section). By this, we have

$$\begin{aligned} t'_1 &\triangleq t'_j, t'_2 \triangleq t'_a, t'_3 \triangleq t'_j + t'_a, t'_4 \triangleq t'_v, \\ t'_5 &\triangleq t'_j + t'_v, t'_6 \triangleq t'_a + t'_v, t'_7 \triangleq t'_j + t'_a + t'_v. \end{aligned} \quad (13)$$

The generated discrete-time acceleration profile sequence is

$$\ddot{x}^e(n) = \begin{cases} J_{\lim}(nt_{sp}) & \text{for } 0 \leq nt_{sp} \leq t'_1 \\ J_{\lim}t'_1 & \text{for } t'_1 \leq nt_{sp} \leq t'_2 \\ -J_{\lim}(nt_{sp} - t'_3) & \text{for } t'_2 \leq nt_{sp} \leq t'_3 \\ 0 & \text{for } t'_3 \leq nt_{sp} \leq t'_4 \\ -J_{\lim}(nt_{sp} - t'_4) & \text{for } t'_4 \leq nt_{sp} \leq t'_5 \\ -J_{\lim}t'_1 & \text{for } t'_5 \leq nt_{sp} \leq t'_6 \\ J_{\lim}(nt_{sp} - t'_7) & \text{for } t'_6 \leq nt_{sp} \leq t'_7 \end{cases} \quad (14)$$

where $x^e(\cdot)$ is the discrete-time profile sequence, $t'_i (i = 1, \dots, 7)$ is the rounded-off switching

times, and n is the serial number of the sampling time instance.

Obviously, the relation between the rounded-off profile eigenvalues x'^d , $J'_{\max}$, $a'_{\max}$, $v'_{\max}$ and the rounded-off switching times still holds as (9)-(12), and can be given as

$$J'_{\max} = J_{\lim}, \quad (15)$$

$$a'_{\max} = J_{\lim} \cdot t'_j, \quad (16)$$

$$v'_{\max} = J_{\lim} \cdot t'_j \cdot t'_a, \quad (17)$$

$$x'^d = J_{\lim} \cdot t'_j \cdot t'_a \cdot t'_v. \quad (18)$$

Here, the calculations of (15)-(18) are omitted.

The mathematical model of DEEP is given as

$$\Delta x^d = (t'_j t'_a t'_v - t_j t_a t_v) J_{\lim} \quad (19)$$

$$\cdot = \left(t_{sp}^3 \left[\frac{t_j}{t_{sp}} \right] \left[\frac{t_a}{t_{sp}} \right] \left[\frac{t_v}{t_{sp}} \right] - t_j t_a t_v \right) J_{\lim}$$

where $\lceil \cdot \rceil$ denotes the ceil-round-off function. It can be seen from (19) that DEEP is proportional to $J_{\lim}$, but has a complicated nonlinear property versus t_j , t_a , t_v and t_{sp} due to the nonlinear ceil-round-off function.

Adaptive Correcting of DEEP

To satisfy the discrete-time endpoint position accuracy, we propose an adaptive method to correct DEEP by introducing a single adaptive factor ξ . This factor ξ should satisfy

$$\xi = \prod_{i=j,a,v} \left(\frac{t_i}{t'_i} \right), \quad (i = j, a, v). \quad (20)$$

By this factor, the acceleration profile sequence can be adaptively scaled as

$$\ddot{x}^c(n) = \ddot{x}^e(n) \times \xi, \quad (21)$$

where $x^c(\cdot)$ is the corrected discrete-time profile sequence. Then the discrete-time velocity and position profile sequences can be calculated by forward-direction integrating (21), as

$$\dot{x}^c(n) = \sum_{k=0}^{n-1} (\ddot{x}^c(k) t_{sp}), \quad (22)$$

$$x^c(n) = \sum_{k=0}^{n-1} (\dot{x}^c(k) t_{sp}). \quad (23)$$

In the following, two propositions are given to proof that the obtained corrected discrete-time profile sequences (21)-(23) satisfy the objective and constraints given in Section-II.

Proposition 1: The corrected DEEP notated as $\Delta^* x^d$ is ideal zero, as $\Delta^* x^d = 0$.

Proof 1: From (21), we have the corrected target distance

$$x^{*d} = x^c(n_f) = \xi x^e(n_f) = \xi x'^d, \quad (24)$$

1. Continuous-time planning

- 1.1. Set target distance and physical limits.
- 1.2. Calculate three critical time durations as shown in Figure 1 according to [1], i.e. max jerk duration t_j , max acceleration duration t_a , and max velocity duration t_v . The switching time can be determined as $t_1 \triangleq t_j$, $t_2 \triangleq t_a$, $t_3 \triangleq t_j + t_a$, $t_4 \triangleq t_v$, $t_5 \triangleq t_j + t_v$, $t_6 \triangleq t_a + t_v$, $t_7 \triangleq t_j + t_a + t_v$

2. Discrete-time acceleration profile sequence generating

- 2.1. Ceil-round-off t_j , t_a , t_v to t'_j , t'_a , t'_v and keep the S-curve's symmetry nature in discrete-time domain as $t'_1 \triangleq t'_j$, $t'_2 \triangleq t'_a$, $t'_3 \triangleq t'_j + t'_a$, $t'_4 \triangleq t'_v$, $t'_5 \triangleq t'_j + t'_v$, $t'_6 \triangleq t'_a + t'_v$, $t'_7 \triangleq t'_j + t'_a + t'_v$
- 2.1. Calculate the discrete acc. profile sequence:

$$\ddot{x}^e(n) = \begin{cases} J_{\lim}(nt_{sp}) & \text{for } 0 \leq nt_{sp} \leq t'_1 \\ J_{\lim}t'_1 & \text{for } t'_1 \leq nt_{sp} \leq t'_2 \\ -J_{\lim}(nt_{sp} - t'_3) & \text{for } t'_2 \leq nt_{sp} \leq t'_3 \\ 0 & \text{for } t'_3 \leq nt_{sp} \leq t'_4 \\ -J_{\lim}(nt_{sp} - t'_4) & \text{for } t'_4 \leq nt_{sp} \leq t'_5 \\ -J_{\lim}t'_1 & \text{for } t'_5 \leq nt_{sp} \leq t'_6 \\ J_{\lim}(nt_{sp} - t'_7) & \text{for } t'_6 \leq nt_{sp} \leq t'_7 \end{cases}$$

3. Adaptive correcting of DEEP

- 3.1. Introduce a single correction factor: $\xi = \prod_{i=j,a,v} (t_i/t'_i)$, $(i = j, a, v)$
- 3.2. Scales down the acceleration for correction: $\ddot{x}^c(n) = \ddot{x}^e(n) \times \xi$
- 3.3. Calculate the correct vel. and pos. profile sequence: $\dot{x}^c(n) = \sum_{k=0}^{n-1} (\ddot{x}^c(k) t_{sp})$, $x^c(n) = \sum_{k=0}^{n-1} (\dot{x}^c(k) t_{sp})$

4. Result in zero DEEP & limits satisfactory

FIGURE 4. DEEP correcting algorithm.

where n_f is the final sampling time instance. Substituting (18) and (20) into (24) leads to

$$x^{*d} = J_{\lim} \times \prod_{i=j,a,v} \left(t'_i \times \frac{t_i}{t'_i} \right) = x^d. \quad (25)$$

Hence, we have

$$\Delta^* x^d = |x^{*d} - x^d| = 0. \quad (26)$$

Proposition 2: Suppose that the critical time durations t_j , t_a , t_v are all ceil-rounded-off as

$$t'_i = \left\lceil \frac{t_i}{t_{sp}} \right\rceil \times t_{sp}, \quad (i = j, a, v), \quad (27)$$

the corrected discrete-time profile sequences strictly satisfy the physical limits.

Proof 2: The proof is similar to the proof of Proposition 1 and is omitted here.

Discussion

The full S-curve profile generation method with DEEP correction is summarized in Figure 4. The given three-step procedure leads to an adaptive but precise correction of the endpoint position accuracy loss and the given physical limits are strictly satisfied. It also can be seen that the correcting procedure is quite simple and would be suitable for real-time industrial applications.

NUMERICAL EXPERIMENTAL RESULTS

To verify the effectiveness of the proposed DEEP correction algorithm, four configurations of parameters, which correspond to the four cases in Figure 3, are given in Table 1. The sampling period is 0.4ms, and the used floating point is 64-bit.

TABLE 1. Configurations of parameters.

Conf.	Jerk limit $J_{lim}(m/s^3)$	Acc. limit $a_{lim}(m/s^2)$	Vel. limit $v_{lim}(m/s)$	Target distance $x^d(mm)$
C1	1000.00	7.00	0.10	0.40
C2			0.03	0.50
C3			0.50	20.00
C4			0.18	20.00

TABLE 2. Experimental results of profile eigenvalues

Conf.		C1	C2	C3	C4
pos. error (mm)	orig.	3.200 e-2	2.685 e-2	7.567 e-1	8.166 e-1
	corr.	1.626 e-16	-1.084 e-16	-3.469 e-15	7.633 e-14
endpoint pos. (mm)	ideal	0.4000	0.5000	20.000	20.000
	orig.	0.4320	0.5268	20.757	20.817
	corr.	0.4000	0.5000	20.000	20.000
max. vel. (m/s)	ideal	0.0342	0.0300	0.3505	0.1800
	orig.	0.0360	0.0314	0.3629	0.1872
	corr.	0.0333	0.0298	0.3497	0.1799
max. acc. (m/s ²)	ideal	5.8480	5.4772	7.0000	7.0000
	orig.	6.0000	5.6000	7.2000	7.2000
	corr.	5.5556	5.3146	6.9375	6.9175
max. jerk (m/s ³)	ideal	1000.0	1000.0	1000.0	1000.0
	orig.	1000.0	1000.0	1000.0	1000.0
	corr.	925.93	949.04	963.54	960.77

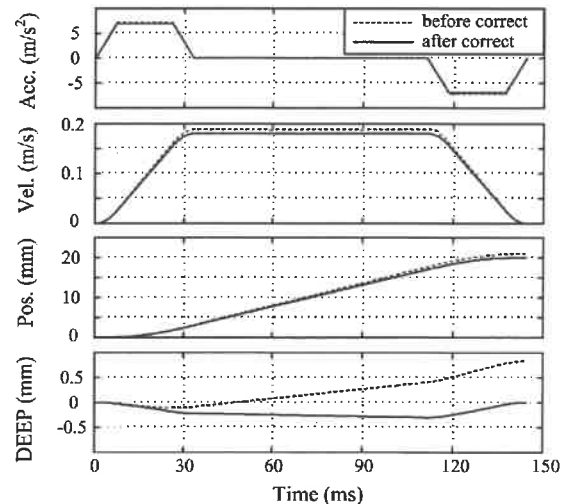


FIGURE 5. Profiles for C4.

Comparative experimental results of the profile eigenvalues under configurations C1-4 are summarized in Table 2, and the motion profiles of C4 is shown in Figure 5 as an example. In all cases, the proposed algorithm greatly improved the accuracy of endpoint position only with the error of less than millionth of a nanometer. And the calculated J_{max} , a_{max} , v_{max} are stay within their given physical limits J_{lim} , a_{lim} , v_{lim} .

CONCLUSION

Summary

In this paper, a method to correct the endpoint position error of reference profile caused by time discretization for the S-curve profile is proposed. By introducing an adaptive factor, the endpoint position error of the reference profile is greatly reduced. Besides, the maximal jerk, the maximal acceleration and the maximal velocity are still stay within the given physical limits. Moreover, the proposed algorithm is simple and adaptive, which is quite suitable for real-time applications.

Future Work

The results from Table 2 show that there is still some endpoint position imprecision that results from the calculation quantization error. It will be further considered in our future work.

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HYBRID CONTROLLER TO COMPENSATE FOR FRICTIONS IN BALLSCREW FEEDDRIVES AT LOW AND HIGH SPEED TRACKING

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ABSTRACT

Mechanical property of a feed drive system subjected to friction effect is sensitive to their operating conditions. The static and dynamic friction models are required to reject undesirable effects by feedforward methods. Inaccurate friction models degrade tracking performance as the feed drive system has time-variant characteristics. In this paper, a hybrid controller is proposed to improve tracking performance. It is composed of the friction model based feedforward compensator and the modified Kalman filter with the inner loop. The control performance has been verified through experiments.

INTRODUCTION

Precision machinery equipped with ballscrews is widely used in the manufacturing, machine tool and semiconductor industries. Friction is very nonlinear at velocity reversals. Tracking errors due to friction deteriorate positioning accuracy regarding operating speeds. At low speed operation, the friction force is a hysteresis function of position in the presliding regime. For high speed motion, the friction is a nonlinear function of velocity in the sliding regime.

To compensate for friction forces, model-based feedforward compensators, adaptive control, disturbance observers (DOB) and repetitive control have been studied in linear servos [1~3]. For good feedforward control using the control structure shown in Fig. 1, exact estimation of friction is required first. However, inaccurate friction estimation degrades control performance. Unmodeled friction components and the hysteresis change degenerate the control performance as well. An inner loop controller with torque estimators shown in Fig. 2 was applied for the disturbance rejection [4].

HYBRID CONTROLLER

At very low speed, friction of the presliding regime depends upon junction deformation. It consists of time varying hysteresis loop. Dynamic friction compensation through the feedforward controller is required. On the other hand, Coulomb and viscous frictions are dominant at the high speed region. Static friction model with Coulomb, viscous and Stribeck effects should be identified accurately. Then, they should be compensated completely through the controller in the low and high speed regions.

In this paper, the static and dynamic friction models are identified at the whole speed range. Generalized Maxwell-slip model [5] is used for the dynamic friction model. A hybrid controller composed of the model based feedforward friction compensator and the model free internal loop controller linked with the modified Kalman filter shown in Fig. 3 is proposed for the friction compensation in ballscrew servos. For accurate identification of friction models at low and high speed regions effectively, a special experimental setup shown in Fig. 4 have been fabricated. An automatic friction parameter gathering linked with the hybrid controller is devised through interface between the xPC target and the experimental setup. In addition, accurate servo modeling is conducted for better friction identification. Performance of the proposed hybrid friction controller is verified at low/high-speed tracking and velocity reversals.

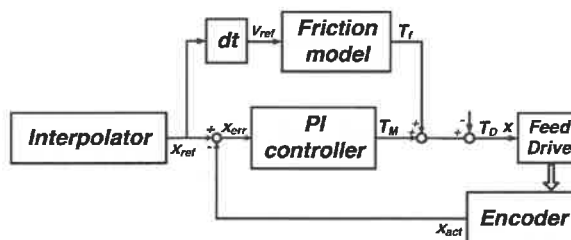


FIGURE 1. Model based feedforward friction control system.

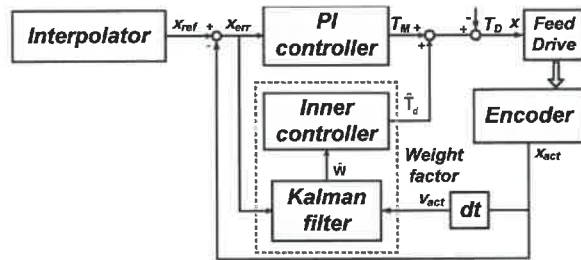


FIGURE 2. Observer based inner loop friction control system.

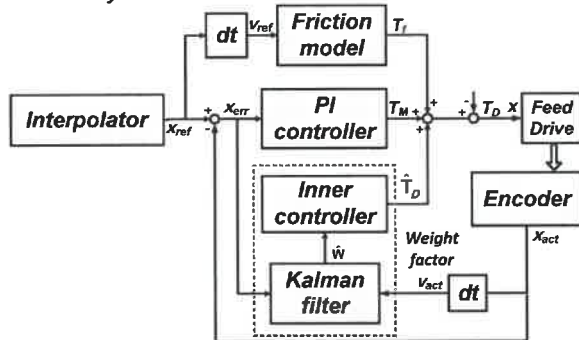


FIGURE 3. Hybrid friction control system with modified Kalman filter.

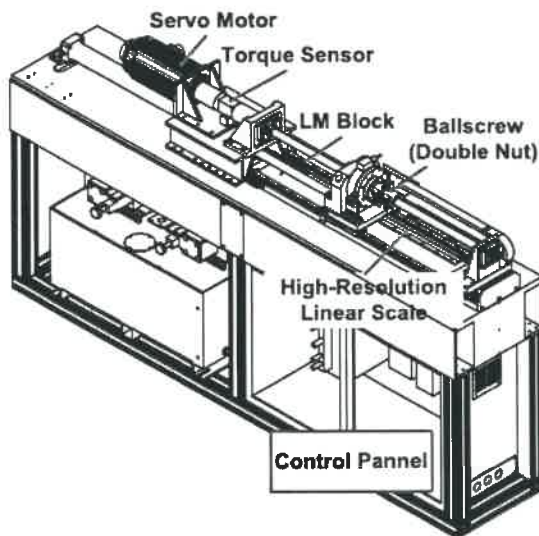


FIGURE 4. Experimental setup of the ballscrew servo.

EXPERIMENTAL SETUP

Fig. 4 shows a ballscrew-driven feeddrive servo. Torque sensor is mounted between the servo motor and the ballscrew to measure friction torque generated at the feeddrive. Double-nut type ballscrew with shaft diameter 36mm and lead 10mm is used for experiments. Linear

encoder with 20nm resolution is installed to observe dynamic characteristics in presliding regime motion. Real-time xPC Target is interfaced to control the experimental setup as shown in Fig. 5.

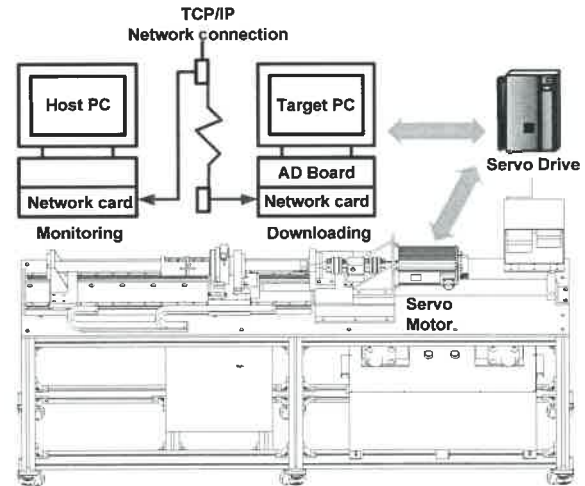


FIGURE 5. xPC-target based servo system.

MODIFIED KALMAN FILTER COMPENSATOR

In the position control servo, tracking error comes from friction torque. It is modeled as a disturbance. Using state variables, a discrete state equation is obtained as follows:

$$\begin{aligned} x_{k+1} &= Ax_k + w_k \\ z_k &= Hx_k + \kappa_k \end{aligned} \quad (1)$$

Defining the tracking error and the estimated friction torque observed through the Kalman filter, following state equation is obtained

$$\begin{bmatrix} Err \\ \hat{T} \end{bmatrix}_{k+1} = \begin{bmatrix} 1 & \Delta t \\ B_{eq} & M_{eq} \end{bmatrix} \begin{bmatrix} v \\ \alpha \end{bmatrix}_k + \begin{bmatrix} 0 \\ w \end{bmatrix}_k \quad (2)$$

where Err , $\hat{T}$, v , α and w are tracking error, estimated torque, velocity, acceleration and white noise, respectively. Δt , B_{eq} and M_{eq} are sampling time, equivalent viscous and mass of the system, respectively. Measured output is given by

$$z_k = \begin{bmatrix} 1 & 0 \end{bmatrix} \begin{bmatrix} v \\ \alpha \end{bmatrix}_k + \kappa_k \quad (3)$$

where z and κ are measured velocity and noise.

In order to observe frictional torque of the servo, modified Kalman filter (MKF) [6] is applied to estimate the friction torque as follows:

$$\begin{aligned}\hat{x}_k^- &= A\hat{x}_{k-1} \\ P_k^- &= AP_{k-1}A^T + Q \\ K_k &= P_k^- H^T (HP_{k-1}H^T + R)^{-1} \\ \hat{x} &= \hat{x}_k^- + K_k (z_k - H\hat{x}_k^-) \\ P_k &= P_k^- - K_k HP_k^-\end{aligned}\quad (3)$$

where P and K are error covariance matrix and Kalman gain. Superscripts ' $\wedge$ ' and ' $-$ ' are estimated and predicted values.

FRICTION MODELS

Friction occurs from relative motion between two surfaces and has strong nonlinearity. It divides into the static friction generated at the sliding regime and the dynamic friction generated at the presliding regime.

Static Friction Model

Static friction model is described by a function of velocity in the sliding regime as follows:

$$F(v) = \text{sign}(v) \cdot \left(F_c + (F_s - F_c) \exp\left(-\left|\frac{v}{V_s}\right|^\delta\right) \right) + F_v \cdot v \quad (4)$$

where F_c , F_s , V_s , δ and F_v are Coulomb friction, static friction, Stribeck velocity, Stribeck shape factor and viscous friction coefficients, respectively.

Generalized Maxwell-Slip Model (GMS)

Lampaert et al. [5] proposed the Generalized Maxwell-Slip (GMS) friction model. It is a further development of the Leuven model to account for hysteresis function in the presliding regime. It is composed of Stribeck curve for constant velocity, hysteresis function with non-local memory in the pre-sliding regime and frictional memory in the sliding regime. Its friction force is given by

$$F_f(t) = \sum_{i=1}^N F_i(t) + F_v v(t) \quad (5)$$

Elementary friction force of a slip-block F_i behaves a dynamic spring model with stiffness

during slipping and sticking motion. F_v means viscous term. During sticking, elementary friction force is given by

$$\frac{dF_i}{dt} = k_i v \quad (6)$$

In the sliding regime, F_i is given by

$$\frac{dF_i}{dt} = \text{sgn}(v) \cdot C \left(\alpha_i - \frac{F_i}{S(v)} \right) \quad (7)$$

α_i is the normalized sustainable maximum friction force in the pre-sliding regime, $S(v)$ is the Stribeck curve, and C is the constant parameter following the Stribeck effect in the sliding regime.

FRICTION MODEL IDENTIFICATION

To identify parameters of the friction models, static experiment is performed at each constant velocity as shown in Fig. 6. Experiments for GMS model are conducted by using a sinusoidal signal with 0.025Hz and amplitude of 0.007mm. Fig. 7 shows measured data and fitting results for the GMS model. Identified parameters are shown in Tables 1 and 2.

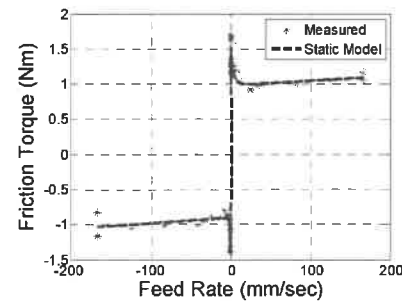


FIGURE 6. Measured friction and static model.

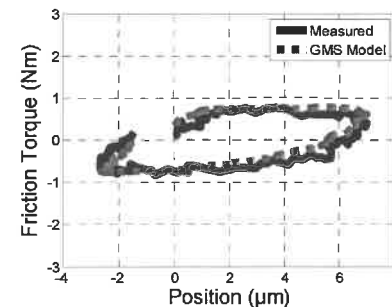


FIGURE 7. Measured friction and GMS model.

EXPERIMENTAL RESULTS

Sinusoidal signals with frequency 0.025Hz and amplitude of 1, 0.1 and 0.05mm are applied to verify control performance as reference inputs. No compensation (NC), GMS friction compensation (GMSC), modified Kalman filter compensation (MKFC), and hybrid control models are applied for comparisons. Figures 8 ~ 10 show experimental results for NC, GMSC and MKFC cases. Among them, the GMSC and MKFC provides good tracking performance at slow motions where the presliding friction is dominant. Fig. 11 shows experimental results for hybrid controller. Using both feedforward and inner loop control with GMS friction model, tracking performance has better performance than other controllers in slow and high speed motions. Table 3 shows summary of friction compensation results.

TABLE 1. Static model: friction parameters.

	Value	
	Positive	Negative
F_c	0.9897 Nm	0.9007 Nm
F_s	1.5204 Nm	1.3888 Nm
F_v	0.00067 Nm	0.00078 Nm
V_s	2.8511 m/s	0.5826 m/s
δ	0.7497	1.6307

TABLE 2. GMS model: friction parameters.

I	Value		Value
k_1	772.06 N/m	α_1	0.594
k_2	520.38 N/m	α_2	0.400
k_3	81.30 N/m	α_3	0.063
k_4	7.90 N/m	α_4	0.006
C	400 Nm/s ⁻¹		

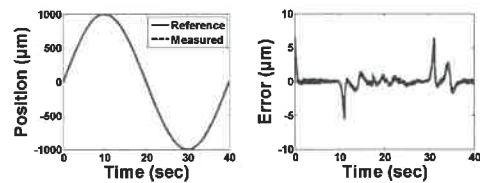
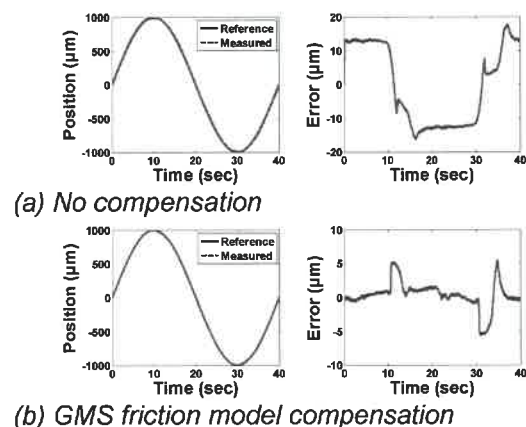


FIGURE 8. Tracking performance regarding $\sin(\pi t / 20)$ (mm) reference input.

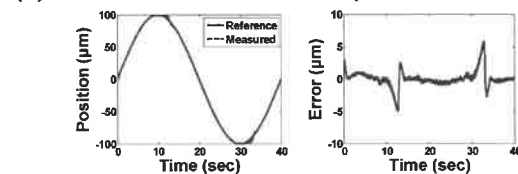
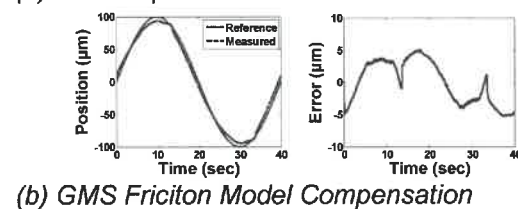
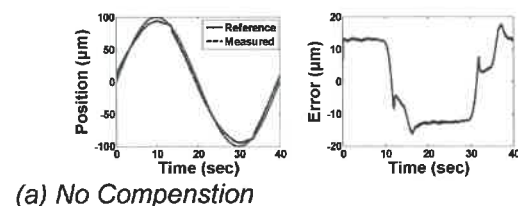


FIGURE 9. Tracking performance regarding $0.1 \sin(\pi t / 20)$ (mm) reference input.

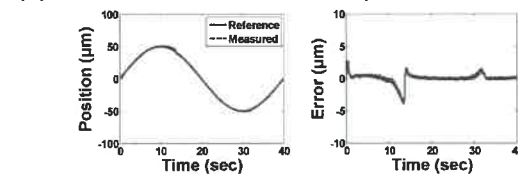
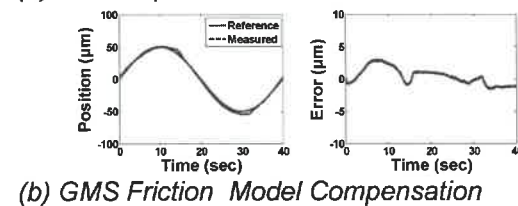
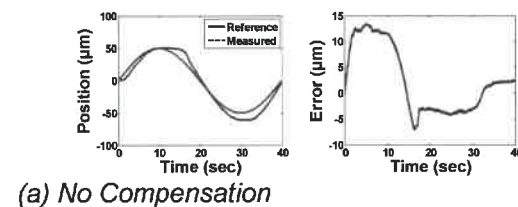
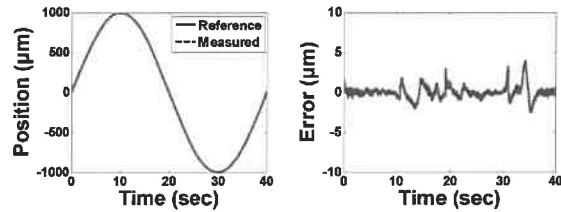
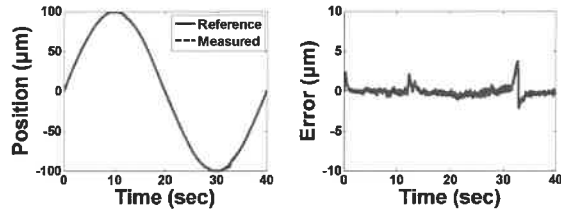


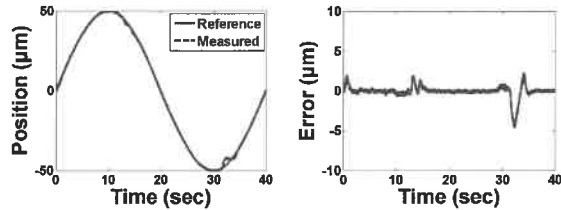
FIGURE 10. Tracking performance regarding $0.05 \sin(\pi t / 20)$ (mm) reference input.



(a) for $\sin(\pi t / 20)$ (mm) reference input.



(b) for $0.1 \sin(\pi t / 20)$ (mm) reference input.



(c) for $0.05 \sin(\pi t / 20)$ (mm) reference input.

FIGURE 11. Hybrid controller performances.

CONCLUSIONS

Hybrid controller combining the GMS friction model feedforward and the inner loop MKFC has been designed and tested on the ballscrew driven servo. By using dynamic and static friction compensation in the presliding and sliding regimes through the hybrid controller, better tracking performance in slow and high speed motions has been obtained than other feedforward and feedback controllers. Using the inner loop control characteristics of the MKFC, uncertainty due to inaccurate modeling of the servo and frictions has been improved as well. Experiments have confirmed the tracking performance of the devised hybrid controller.

TABLE 3. Comparisons of maximum tracking errors. (Unit: μm)

Amplitude	1000	100	50
NC	24	34	20
GMSC	11	10	9
MKFC	11	10	6
Hybrid Controller	8	5	3

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IDENTIFICATION OF STATIC AND DYNAMIC FRICTIONS IN BALLSCREW SERVOS

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ABSTRACT

Frictions in mechanical systems deteriorates tracking performance. This friction induces unwanted oscillation such as hunting, steady-state errors and limit cycles. To reduce and remove these unstable effects, accurate friction models including both static and dynamic characteristics are required for feedforward compensators. In this paper, accurate identification processes of static and dynamic friction models such as Dahl, LuGre and Generalized Maxwell-Slip (GMS) are described. Experiments are performed on ballscrew servos.

INTRODUCTION

In precision servos, friction is described with presliding and sliding regimes. Adhesive force is dominant at junctions in the presliding regime. Friction depends upon displacement rather than velocity. As the displacement increases, all asperity junctions are broken. Friction force becomes a function of sliding velocity. A static friction model composed of Coulomb, viscous and Stribeck frictions applies to the sliding regime. A dynamic friction model consists of stick-slip, friction lag, break-away, presliding and Stribeck effects. Generalized Maxwell-Slip (GMS), Dahl and LuGre models have used for describing accurate friction of servos [1-4].

Friction in machine tool feeddrives and precision machinery generates unwanted errors, hunting and limit cycles. To reduce friction effect, friction force should be modeled accurately and then be compensated with feedforward and/or internal loop controller. In this paper, accurate and quick identification methodology of static and dynamic frictions in the ballscrew servo is studied.

FRICTION MODELS

Parameters required for the static and dynamic friction models in the ballscrew driven feeddrive system are as follows:

Static model

Static model describes steady-state behavior in the sliding regime, and only depends on the velocity v . Friction force F_f is given by

$$F_f(v) = \text{sign}(v) \cdot \left(F_c + (F_s - F_c) \exp\left(-\left|\frac{v}{V_s}\right|^\delta\right) \right) + \sigma_2 \cdot v \quad (1)$$

where F_c , F_s , V_s , δ and σ_2 are Coulomb friction, static friction, Stribeck velocity, Stribeck shape factor and viscous friction coefficients, respectively.

Dahl model

Eq. (2) is the Dahl model [3] for representing junction deflections in the presliding regime.

$$\frac{dF_f}{dx} = \sigma \left| 1 - \frac{F_f}{F_c} \text{sgn}(\dot{x}) \right|^n \text{sgn} \left(1 - \frac{F_f}{F_c} \text{sgn}(\dot{x}) \right) \quad (2)$$

where F_f , σ , F_c and n are friction force, initial stiffness at the velocity reversal, Coulomb friction and shape of hysteresis, respectively.

LuGre model

Canudas de Wit et al. [4] added average deformation of contacting bristles at the junction to the Dahl model. It combines the presliding behavior of the Dahl model with steady-state friction characteristics in the sliding regime. Friction force as a function of the state variable z and the velocity v is given by

$$F_f = \sigma_0 z + \sigma_1 \frac{dz}{dt} + \sigma_2 v \quad (3)$$

where F_f , σ_0 , σ_1 and σ_2 are friction force, average bristle stiffness, micro-viscous friction

coefficient and viscous friction coefficient, respectively. Average deflection of the bristles z is given by

$$\frac{dz}{dt} = v - \sigma_0 \frac{|v|}{s(v)} z \quad (4)$$

At steady state, the Stribeck effect is given by

$$s(v) = F_c + (F_s - F_c) \exp\left(-\left|\frac{v}{V_s}\right|^\delta\right) \quad (5)$$

where F_c , F_s , V_s and δ are Coulomb friction, static friction, Stribeck velocity and Stribeck shape factor, respectively.

Generalized Maxwell-Slip model

Al-Bender et al. [1] proposed the generalized Maxwell-Slip (GMS) friction model given by

$$F_f(t) = \sum_{i=1}^N F_i(t) + \sigma_2 v(t) \quad (6)$$

Friction force is formulated as a summation of N elementary state models plus an extra viscous term. Dynamic behavior of the GMS model in the presliding region is given by

$$\frac{dF_i}{dt} = k_i v \quad (7)$$

In the sliding region, the state equation is given by

$$\frac{dF_i}{dt} = \text{sgn}(v) \cdot C \left(\alpha_i - \frac{F_i}{S(v)} \right) \quad (8)$$

BALLSCREW SERVO

Fig. 1 shows a ballscrew servo for parameter identification of the friction models. Fig. 2 shows a schematic diagram of the experimental setup. Driving torque of the servo motor is measured through the servo pack. High-resolution 20nm resolution linear encoder is applied for hysteretic behavior measurement in the presliding regime.

FRICTION IDENTIFICATION EXPERIMENTS

Friction depends upon surface asperity, materials of the contact bodies, displacement and relative velocity of the bodies, and lubricant.

To measure and identify static and dynamic friction models, two kinds of experiment called the constant velocity and the breakaway measurements are conducted on the ballscrew servo.

Breakaway Experiment

Static friction is the friction when two bodies are sticking. Force required to overcome the static friction and initiate motion is called the breakaway force. To identify this sticking phenomenon in ballscrew servos, breakaway experiments are required by selection of the velocity control mode first. Then, to eliminate influence of the dwell time effect, a warm up procedure is performed for a minute. After the warm up procedure, the servo pack is switched to the torque control mode. Then, the breakaway torque is measured in the open loop state where a ramp force is applied to the rest servo system.

Fig. 3 shows measured breakaway torque and presliding behavior when the applied torque level is less than 2 Nm. To measure dynamic friction terms in the presliding regime, sinusoidal excitation is required below the breakaway torque level. To minimize inertia effect and to obtain a hysteretic function of displacement, the frequency and amplitude of the sinusoidal input is applied carefully. Fig. 4 shows a measured hysteretic friction and displacement behavior in the presliding regime.

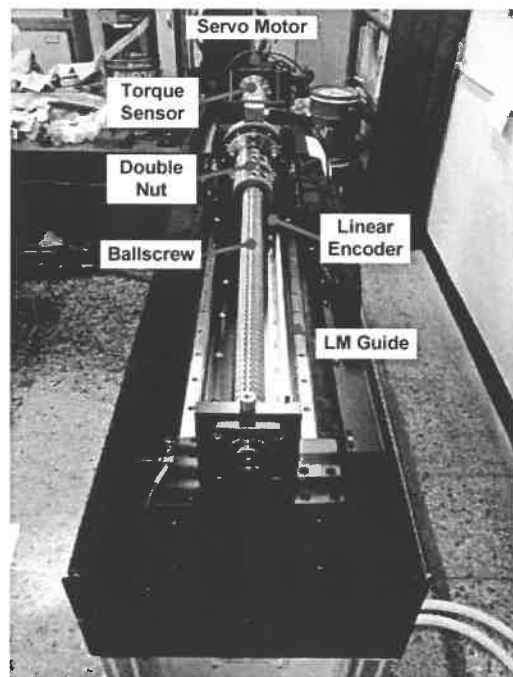


FIGURE 1. Ballscrew-driven feeddrive servo.

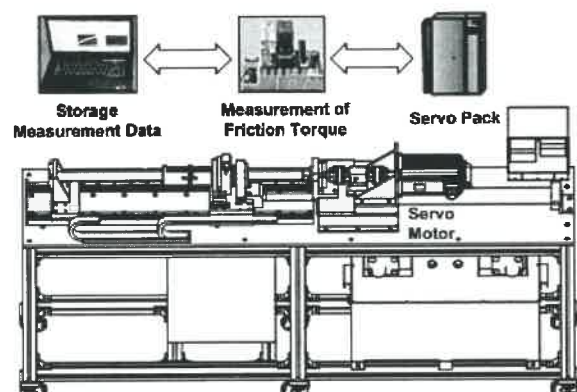
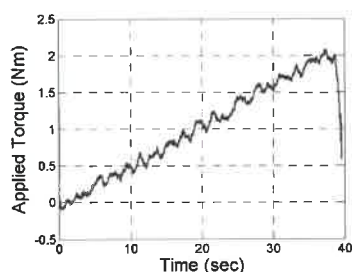
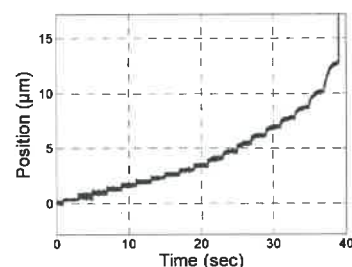


FIGURE 2. Experimental setup.



(a) Applied torque for ramp input.



(b) Position regarding ramp input.

FIGURE 3. Breakaway input and response.

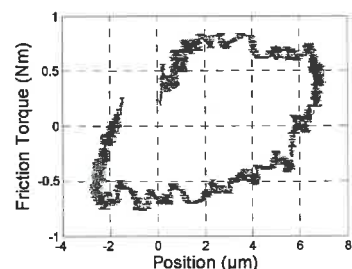


FIGURE 4. Measured hysteresis curve.

Constant Velocity Experiment

A constant velocity experiment is performed to identify parameters of the static friction model during the constant velocity motion. Control mode of the servo pack is selected to the velocity control mode. Then, the servo system is driven to measure friction torque generated at

the servo motor during each constant velocity input from low to high speed.

IDENTIFICATION RESULTS OF FRICTION MODELS

Parameters of the static friction model are identified through the constant velocity experiment. Fig. 5 shows the measured and curve fitted static friction model data. Table 1 lists identified static friction model parameters.

In addition, parameters for dynamic friction models are identified from the hysteresis curve. Fig. 6 shows identification results of dynamic friction models with 4 slip-blocks. Table 2 lists identified dynamic friction parameters.

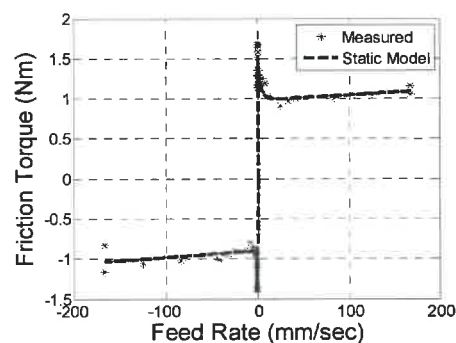


FIGURE 5. Static friction model.

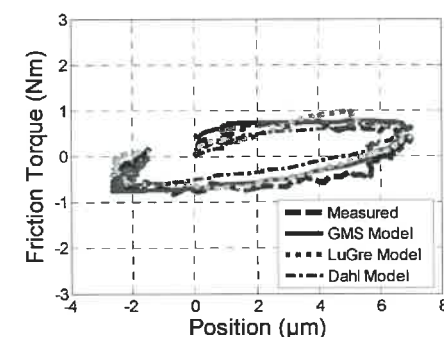


FIGURE 6. Dynamic friction models.

TABLE 1. Static friction model parameters.

Symbol	Value	
	Positive Velocity	Negative Velocity
F_c	0.9897 Nm	0.9007 Nm
F_s	1.5204 Nm	1.3888 Nm
F_v	0.00067 Nm	0.00078 Nm
V_s	2.8511 mm/s	0.5826 mm/s
δ	0.7497	1.6307

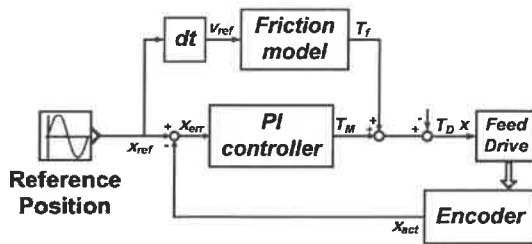


FIGURE 7. Control system with a feedforward friction compensator.

FRICTION COMPENSATION

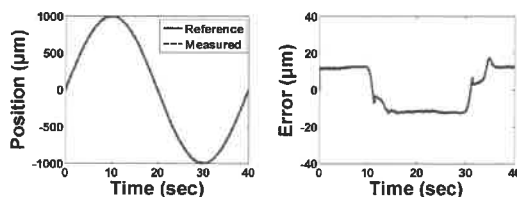
To validate performance of the friction models, sinusoidal inputs with amplitude of 1, 0.1 and 0.05mm, and with frequency of 0.025Hz are applied to the servo system. Fig. 7 shows a feedback control structure with a feedforward friction compensator. Gains of the PI controller are turned manually.

EXPERIMENTAL RESULTS

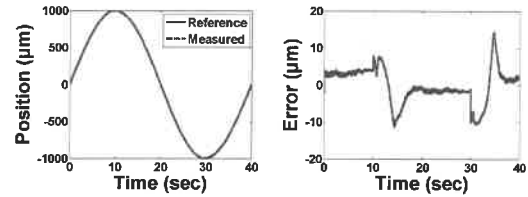
Figures 8-10 show position tracking errors for cases with and without friction feedforward compensators. Using the feedforward friction compensators, better tracking performance has been obtained. Table 3 lists root mean square (RMS) position errors according to each friction model. In case of the static friction model, tracking performance is degraded a lot when the input amplitudes are 0.1 and 0.05mm. These are induced from dynamic friction effect at the presliding regime. It is confirmed that the GMS model shows the best tracking performance.

TABLE 2. Dynamic friction model parameters.

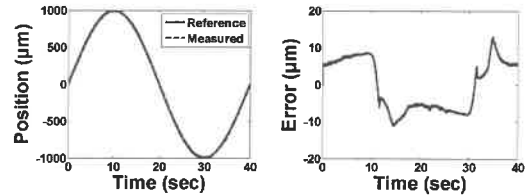
Symbol	Value	Symbol	Value
k_1	772.06 N	α_1	0.594
k_2	520.38 N	α_2	0.400
k_3	81.30 N	α_3	0.063
k_4	7.90 N	α_4	0.006
C	400 N μ m/s	δ	1.720
σ_2	360.28 Ns	σ_0	136 N
σ_1	0.0415 Ns		



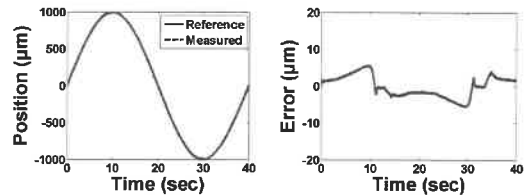
(a) No Friction Compensation



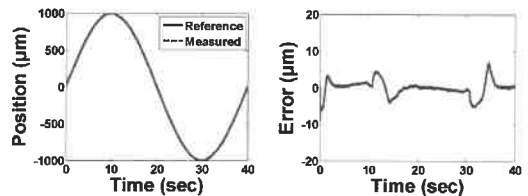
(b) Static Model Compensation



(c) Dahl Model Compensation

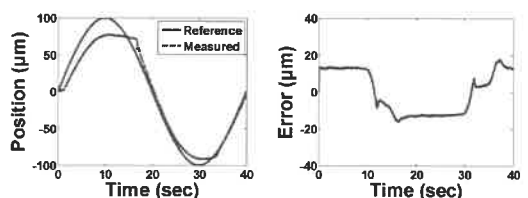


(d) LuGre Model Compensation

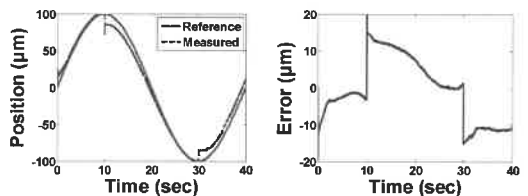


(e) GMS Model Compensation

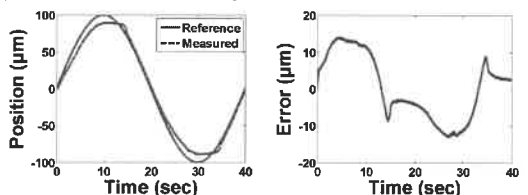
FIGURE 8. Tracking performance regarding $\sin(\pi t/20)$ (mm) reference input.



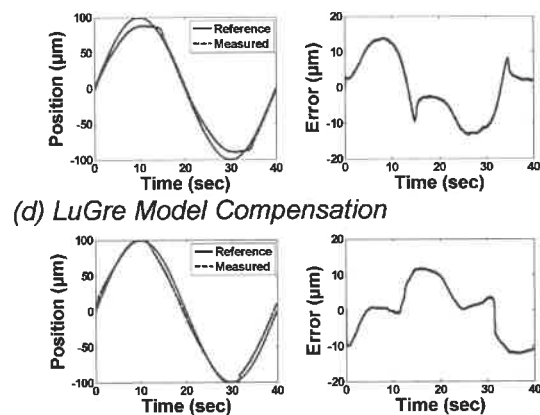
(a) No Friction Compensation



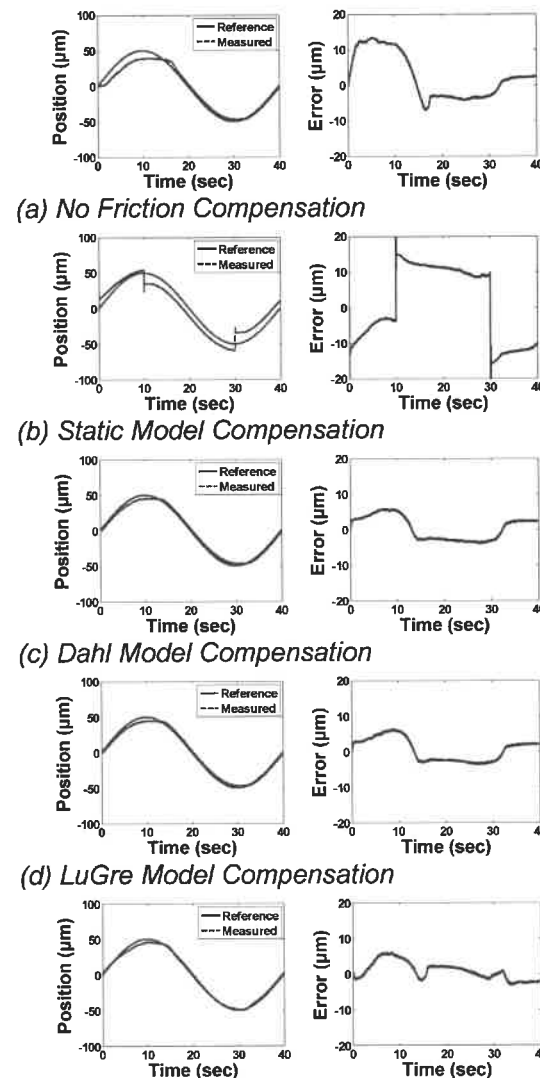
(b) Static Model Compensation



(c) Dahl Model Compensation



(e) GMS Model Compensation
FIGURE 9. Tracking performance regarding $0.1\sin(\pi t/20)$ (mm) reference input.



(e) GMS Model Compensation
FIGURE 10. Tracking performance regarding $0.05\sin(\pi t/20)$ (mm) reference input.

TABLE 3. RMS position tracking errors for 0.1Hz sinusoidal input. (Unit: μm)

Amplitude	1000	100	50
No Compensation	10.92	11.15	5.50
Static Model	3.72	7.19	10.22
Dahl Model	6.63	7.66	3.13
LuGre Model	5.00	7.23	2.98
GMS Model	2.79	5.85	2.16

CONCLUSIONS

Identification procedures of static and dynamic friction models in the ballscrew servo are proposed in this paper. To verify the accuracy of the identification processes, a feedforward friction compensator has been applied to the position tracking servo. In the presliding regime, static model has much error due to the overcompensation of the friction torque. To improve tracking performance at low speed and velocity reversal, dynamic friction models are required. It is confirmed that the GMS model shows the best performance than other friction models.

ACKNOWLEDGEMENT

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DEVELOPMENT OF SIMPLE POWER SUPPLYING SYSTEM IN MULTI-WIRE EDM SLICING METHOD

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INTRODUCTION

Monocrystalline and polycrystalline silicon are widely used in semiconductor and photovoltaic industries. In addition, silicon carbide and gallium nitride have been expected as a material of power semiconductor. These materials are hard and brittle, and it is required to slice these materials efficiently and accurately. A multi-wire saw method is widely applied to slicing of these ingots. However, this method has some problems such as cracks generated on sliced surface and a large kerf loss. On the other hand, a wire EDM method has a possibility to reduce a kerf loss, since a thin wire electrode can be used owing to its small machining force without mechanical contact. In this method, the removal of material can be performed by electrical discharge pulses regardless of material's hardness. Therefore, a multi-wire EDM slicing method was developed to overcome problems of a multi-wire saw method in order to increase the efficiency and the accuracy of slicing with wire EDM technologies [1]. It was reported that a multi-wire EDM slicing method was applied for the manufacturing process of silicon carbide [2, 3], and a multi-slicing with 40 wires was also discussed [4]. Moreover, a wire electrode with a track-shaped section could be expected to reduce a kerf loss even in the case of thin wire electrode [5, 6].

However, a multi-wire EDM slicing equipment becomes very complicated, when many power supplying units are arranged by corresponding the number of processing wire electrode. In an individual power supplying method as shown in Figure 1 (a), the same number of conductivity piece set are required with increasing the number of processing wire electrode. This complicated setup leads to a high initial cost, difficulties of the preparation and a heavy load on the maintenance for a multi-wire EDM slicing equipment.

Therefore, a group power supplying method as shown in Figure 1 (b) is newly proposed to satisfy both a simple equipment and a high efficiency slicing. In this method, discharge pulses to all processing points are supplied by only one conductivity piece set, even if the number of processing wire electrode increased. A resistance of wire electrode between a conductivity piece and a processing point could play as a limiting resistance of EDM power supplying unit to control a peak current of a discharge pulse. In this study, a newly proposed group power supplying method was discussed, and its fundamental characteristics were experimentally investigated.

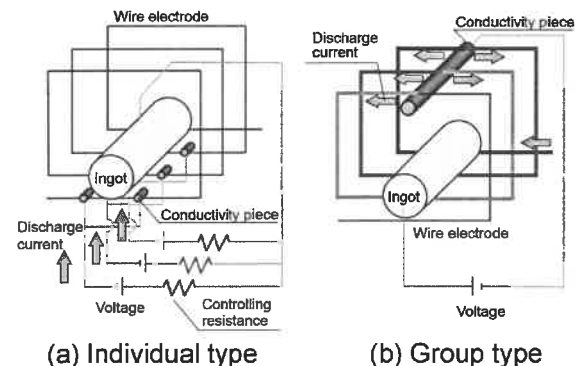


FIGURE 1. Power supplying methods in multi-wire EDM slicing equipment.

EXPERIMENTAL METHOD

Figure 2 schematically shows the wire driving part of the multi-wire EDM slicing equipment. The wire sub guides were prepared at the nearest sides to workpiece to stabilize positions of wire electrodes. Conductivity pieces for supplying discharge pulse to wire electrodes were set at the positions between wire guides and wire sub guides in the case of individual power supplying method. On the other hand, conductivity pieces were set at outer positions of lower wire

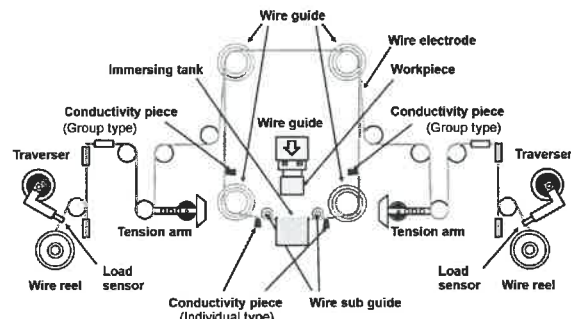


FIGURE 2. Schematic diagram of wire driving part in multi-wire EDM slicing equipment.

TABLE 1. Machining conditions.

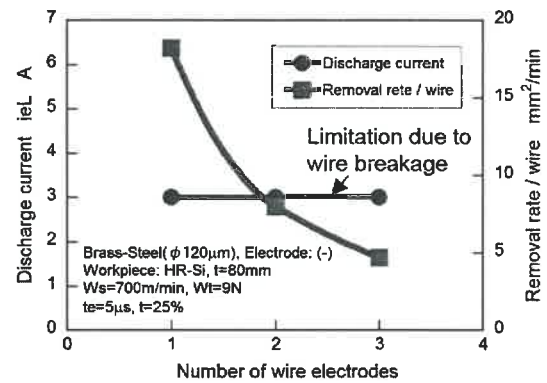
Workpiece	Monocrystalline silicon $\rho_s = 2 - 3 [\Omega \cdot \text{cm}]$
Wire electrode	Brass coated steel $\phi 120 [\mu\text{m}]$
Wire tension	$W_T = 9 [\text{N}]$
Polarity	Electrode [-]
Wire speed	$W_s = 700 [\text{m/min}]$
Open circuit voltage	$ui_H = 300 [\text{V}]$ $ui_L = 120 [\text{V}]$
Discharge duration	$t_e = 5 [\mu\text{s}]$
Duty factor	$\tau = 20 \%$
Dielectric working fluid	Deionized water $\rho_w = 25 [\text{k} \Omega \cdot \text{cm}]$ $\theta_f = 20 [^\circ\text{C}]$

guides to keep a sufficient length of wire electrode in the group power supplying method.

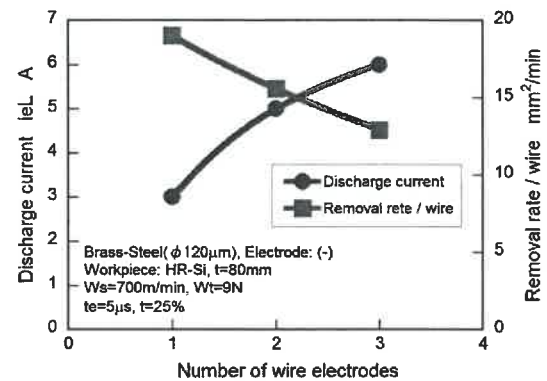
A monocrystalline silicon of $\rho_s = 2 - 3 \Omega \cdot \text{cm}$ in a specific resistance was used as a workpiece, and its width for slicing was 80 mm. Slicing experiments were carried out by moving the workpiece in the downward direction toward the running wires between the wire sub guides.

A brass coated steel wire electrode of 120 μm in diameter was used as a wire electrode. The wire electrode is wound spirally on wire guides approximately 40 times along the wire guide pitch. The wire electrode is fed reciprocally by reversing the feed direction, when the wire electrode comes to the end of set travel length.

Table 1 shows main machining conditions used in this study. A multi-EDM power supply unit can generate lower and longer discharge pulse compared with a normal commercial wire EDM equipment. Also, this electric discharge power supply has a high voltage circuit to start the electrical discharge and a low voltage circuit to generate the electrical discharge. These open



(a) Individual type



(b) Group type

FIGURE 3. Change of removal rate per wire and total peak-discharge-current for with number of wire electrodes.

circuit voltage conditions were expressed as ui_H and ui_L , respectively. Deionized water was used as a working fluid, and its specific resistance was set to $\rho_w = 2.5 \times 10^5 \Omega \cdot \text{cm}$.

RESULTS and DISCUSSION

Figure 3 shows the change of removal rate per one wire electrode and total peak-discharge-current with the number of processing wire electrodes, when one conductivity piece was used with both the individual and the group power supplying methods. In the case of individual power supplying method with one conductivity piece set as shown in Figure 3 (a), the total peak-discharge-current to a workpiece must be constant in order to avoid the breakage of wire electrode. If the total energy increased with increasing the number of processing wire electrodes in order to keep slicing efficiency, a discharge current concentrated in one wire electrode, which resulted in the breakage of wire electrode due to excessive high peak discharge-current. Therefore, the removal rate per one wire

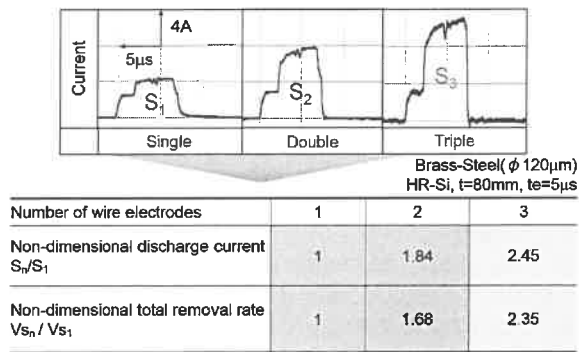


FIGURE 4. Comparison of discharge current waveform area and total removal rate for single, double and triple processing wire electrode slicing by group power supplying method.

electrode can but decrease with increasing the number of processing wire electrodes under a constant peak-discharge-current condition in the individual power supplying method.

On the other hand, in the group power supplying method as shown in Figure 3 (b), the total peak-discharge-current to a workpiece could be increased with increasing the number of processing wire electrodes without the breakage of wire electrode. This phenomenon indicates that the energy of electrical discharge pulse could be homogeneously distributed to each processing wire electrode, although the energy of electrical discharge pulse increased with only one conductivity piece set. Therefore, the group power supplying method could keep similar removal rate with only one conductivity piece set in a multi-wire EDM slicing method, even if the number of processing wire electrodes increased.

Figure 4 shows comparison of discharge current waveform area and total removal rate, when single, double and triple wire EDM slicing were carried out with the group power supplying method. The area of discharge current waveform S_1 and the removal rate V_{S_1} with single wire EDM slicing was defined as a standard value, and the total area of discharge current S_n and the total removal rate V_{S_n} with double and triple wire EDM slicing were divided by that with single wire EDM slicing. Their ratio were calculated in Figure 4. Even in the case of group power supplying method, the removal rate per one wire electrode slightly decreased with increasing the number of processing wire electrodes, as mentioned in Figure 3. Because the total peak-discharge-current could not be increased with similar increasing ratio to the number of processing wire electrodes. When the

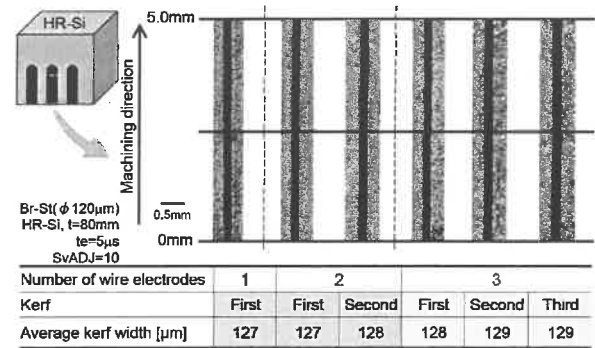


FIGURE 5. Kerfs for single, double and triple wire EDM slicing by group power supplying method.

ratio of total area of discharge current S_n and total removal rate V_{S_n} was compared, the total removal rate V_{S_n} proportionally increased with increasing the total area of discharge current S_n , as shown in Figure 4.

Kerfs for single, double and triple wire EDM slicing are shown in Figure 5, when the group power supplying method was used. As shown in the figure, almost constant kerf widths could be obtained at each processing point, even if the number of processing wire electrodes increased. Normally, a gap distance increases with increasing a total peak-discharge-current in the case of individual power supplying method with one conductivity piece set. On the other hand, the group power supplying method could keep a constant gap distance between a wire electrode and a workpiece, although the total peak-discharge-current to a workpiece increased with one conductivity piece set. This result clarified that input energies were almost constant at each processing wire electrode, and electrical discharge pulses could be homogeneously distributed to each processing wire electrode in the group power supplying method, although the number of conductivity piece set was one for three processing wire electrodes.

Figure 6 shows the comparison of power supplying method in multi-wire EDM slicing with one conductivity piece, when the energy E is suitable electrical discharge energy with sufficient large removal rate without the breakage of wire electrode for one processing wire electrode. In the individual power supplying method as shown in Figure 6 (a), when the total input energy is $3E$, almost all input energy concentrates one processing wire electrode, which is the first start point of electrical discharge. It is considered that the kerf width increases due to a large discharge

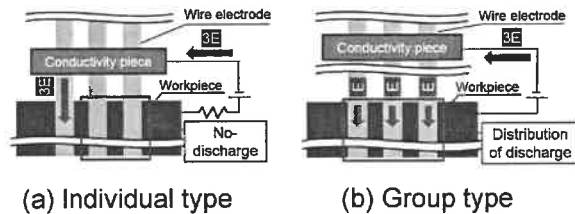


FIGURE 6. Comparison of energy flow by type of power supplying method in multi-wire EDM slicing.

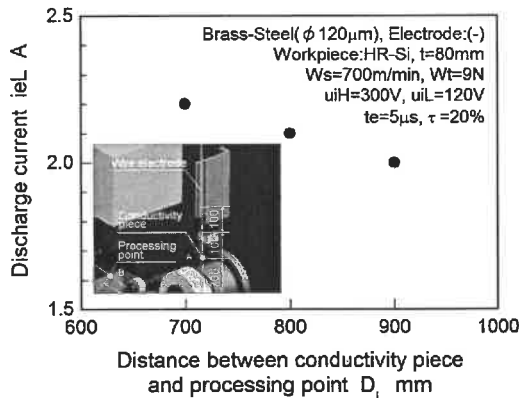


FIGURE 7. Peak-discharge-current and distance between conductivity piece and processing point by group power supplying method.

pulse energy, which leads to the risk of wire electrode breakage. Therefore, it is difficult to keep the similar removal rate per wire electrode in the individual power supplying method with one conductivity piece, when the number of processing wire electrodes increased. On the other hand, the group power supplying method with one conductivity piece can distribute the energy of electrical discharge to each processing wire electrode homogeneously as shown in Figure 6 (b), since the sufficient long wire electrode works as a resistance of electrical discharge circuit. Therefore, it is expected that a similar removal rate per wire electrode could be kept with increasing the energy of electrical discharge pulse by using the group power supplying method with one conductivity piece.

Figure 7 shows the change of peak-discharge-current with the distance between a conductivity piece and a processing point in the group power supplying method. As shown in the figure, the peak-discharge-current proportionally decreased with increasing the distance between a conductivity piece and a processing point. Therefore, the discharge current in the group power supplying method could be controlled by adjusting the length of wire electrode between a conductivity

piece and a processing point. This proportional variation of the peak-discharge-current also indicates that a sufficient long wire electrode could play as a limiting resistance of electrical discharge circuit to control a discharge current in the group power supplying method.

Summary

In this study, a newly proposed group power supplying method was discussed in a multi-wire EDM slicing technique. The total peak current of electrical discharge pulse to a workpiece increased with increasing number of processing wire electrodes without an enlargement of kerf width, and the energy of electrical discharge could be homogeneously distributed to each processing wire electrode without the breakage even by using only one conductivity piece set. The discharge current could be controlled by adjusting the length of wire electrode between the conductivity piece and the processing point. Therefore, it is expected that a newly developed group power supplying method can simplify a construction of multi-wire EDM slicing equipment only by one conductivity piece without a large reduction of removal rate.

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DYNAMIC SIMULATION OF MULTI-AXIS MACHINE TOOLS AND ITS EXPERIMENTAL VERIFICATION

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INSTRUCTIONS

Control and dynamic system simulator gives a chance to system designers to estimate what kind of dynamic performance can be achieved from design parameters and selected devices without doing real experiments. Initially our simulator was built only with common Simulink blocks to model dynamic components in control systems of machinery equipment. To cover a number of commercial mechanical and electrical parts, a set of database for motors, drives, bearings, etc. was also developed and included in our simulator. To describe 6 degrees-of-freedom motion of a moving table, the equations of motion can be derived from Figure 1 by Lagrangian equation.

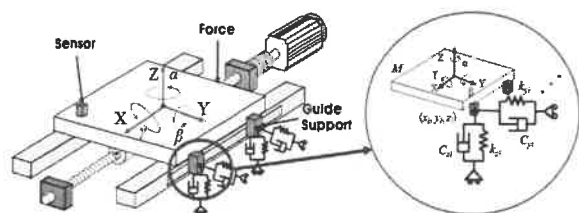


FIGURE 1. General moving table model supported by 4 corner linear guide blocks.

Even though a state-space representation of the 6 d.o.f. equations of motion of the moving table in Simulink is handy, the derivation is lengthy and complicated when it is assumed that sensor and motor locations are arbitrary and there are four horizontal and four vertical spring-damper rollers at each corner of the table as shown in the figure. The situation gets worse when mechanical models of multi-axis machine tools need to be built. Furthermore the maintenance and expandability of the equations of motion are quite poor. Thus, the equations of motion represented by common Simulink blocks were dropped out from our simulator and mechanical parts were modeled by SimMechanics blocks. Designers can model the multibody system using blocks representing bodies, joints, constraints, and force elements without deriving

the equations of motion for the complete mechanical system. Using SimMechanics, we successfully extend the capability of our simulator to handle multi-axis machine tools. As a demonstration, we present a dynamic model and simulation result of a 4-axis machine tool in this paper.

A 4-AXIS MACHINE TOOLS AND ITS MODEL

Figure 2 shows the 4-axis machine tools and its solid model. All axes in the 4-axis machine tool have hydrostatic bearings. The C-axis is driven by a direct drive motor and other axes adopt linear motors.

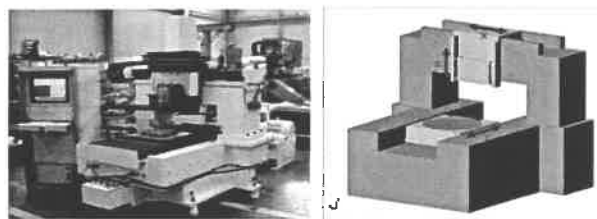


FIGURE 2. A 4-axis machine tools and its mechanical.

To make the moving table have 6 d.o.f. motion, a mechanical guide block in the vertical direction is modeled as shown in Figure 3 using an in-plane joint, a prismatic joint, a spherical joint, and a spring-damper. Note that a moving table has eight sets of blocks shown in Figure 3: four sets for the vertical direction and additional four sets for the horizontal direction. A bushing joint is used to model a rotational axis. The bushing joint has three translational and three rotational degrees and it allows unconstrained, combined 3D translation and rotation.

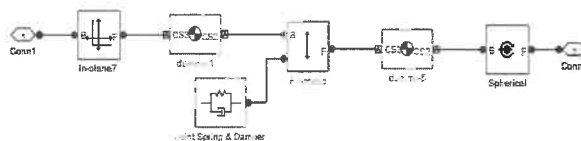


FIGURE 3. Linear guide model using SimMechanics blocks.

When a linear axis modeled by SimMechanics blocks, was compared with the equations of motion represented by a state-space block, it was found that the outputs from two models produced less than 0.01% difference in their values in our test. The SimMechanics model for the 4-axis machine tool's mechanical moving parts is shown at Figure 4. As expected, the Z-axis is stacked on top of the X-axis and the C-axis is on the Y-axis. Other configurations of multi-axis machine tools can be easily modeled in this way. Simple copy-and-paste can produce different multi-axis machine tool models.

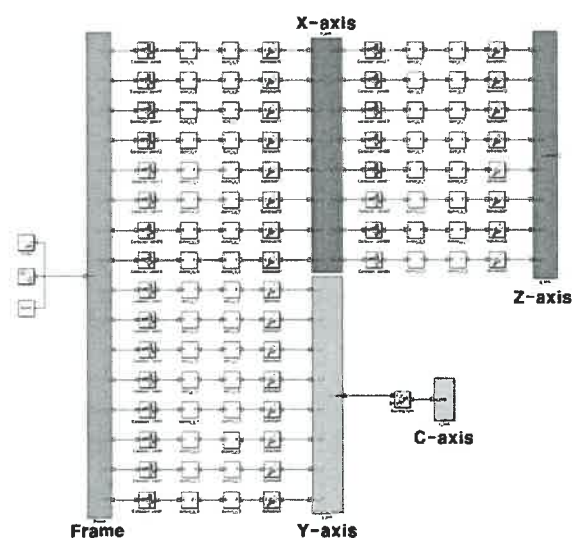


FIGURE 4. Mechanical system model for the 4-axis machine tool using SimMechanics blocks.

Other model blocks for motors, motor drives, and a controller are connected to the SimMechanics multi-axis mechanical model. The whole model of the 4-axis machine tool was simulated and some of its outputs are shown in Figure 5. In Figure 5, the output from the X-axis was compared with the measured experimental data. The simulation results matches well with the experimental data. The circular response consisted of X- and Y-axis motions shows good agreement between the simulation and the experiment.

A COMMERCIAL MACHINE TOOL AND ITS MODEL

To confirm the utility of the modeling method, experiments were carried out with respect to the other machine tool. Figure 6 shows the commercial 3-axis machine tool(Doosan Infracore NX 6500 II) and its solid model. This machine controlled by commercial CNC

controller(Fanuc 31i) and all axes in the machine tool have linear motion guides and servo motors.

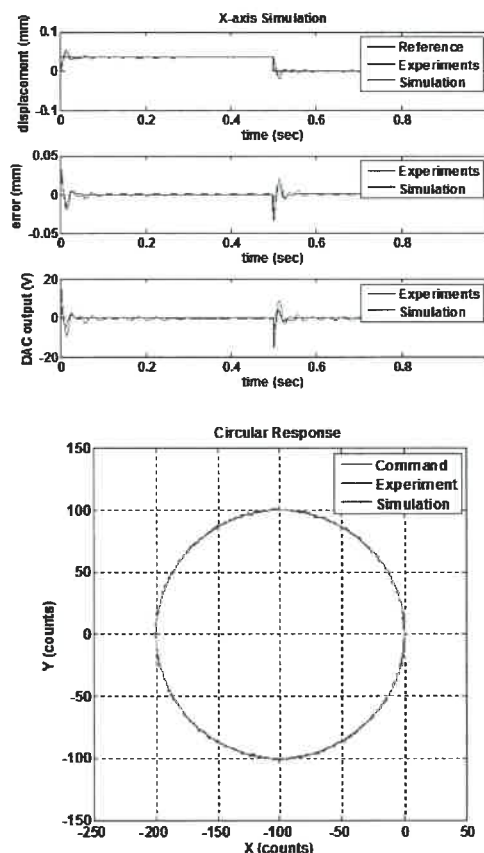


FIGURE 5. Comparison of simulation and experiment results.

The SimMechanics model for the 3-axis machine tool's mechanical moving parts is shown at Figure 7.

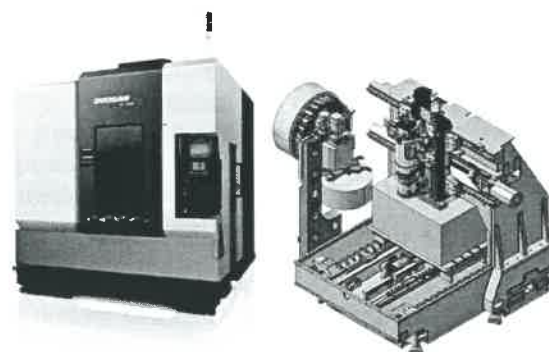


FIGURE 6. Commercial 3-axes machine tool and its solid model.

The whole model of the 3-axis machine tool was simulated and some of its outputs are shown in Figure 8. In Figure 8, the step response from the

X-axis was compared with the measured experimental data. The simulation results matches well with the experimental data.

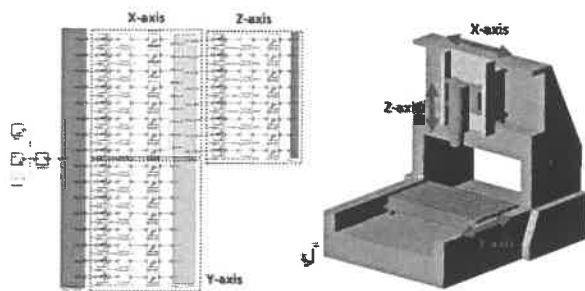


FIGURE 7. Mechanical system model for the commercial machine tool using SimMechanics blocks.

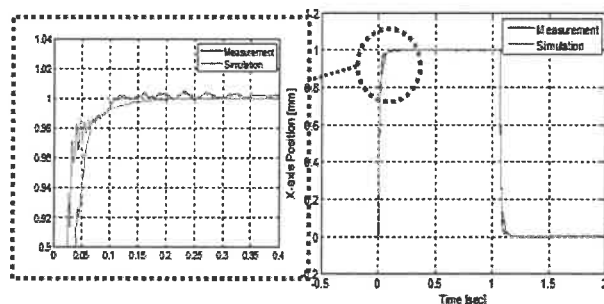


FIGURE 8. Step response comparison of simulation and experiment.

A LEAD SCREW DRIVEN 2-AXIS STAGE MODEL

A 2-axis XY stage which has ball screws and DC servo motors were built and experiments were carried out. Figure 9 and Figure 10 are XY stage and its SimMechanics model, respectively and Figure 11 and 12 are experimental results. The experimental results show that SimMechanics model for the XY stage is quite good matches.

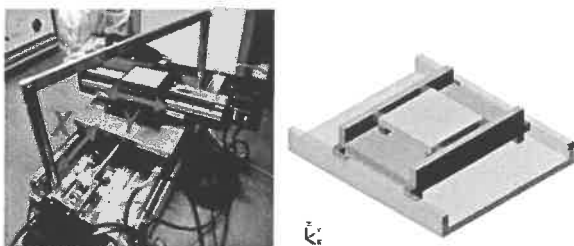


FIGURE 9. Step response comparison of simulation and experiment.

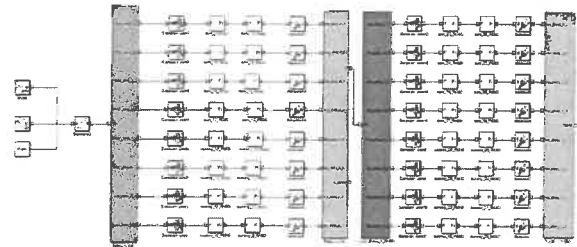
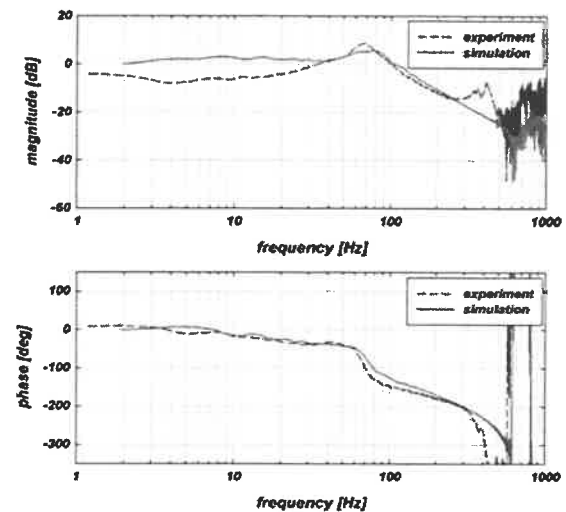


FIGURE 10. SimMechanics Model for the XY stage.



FRF response of Y-axis

FIGURE 11. FRF response comparison of the XY stage.

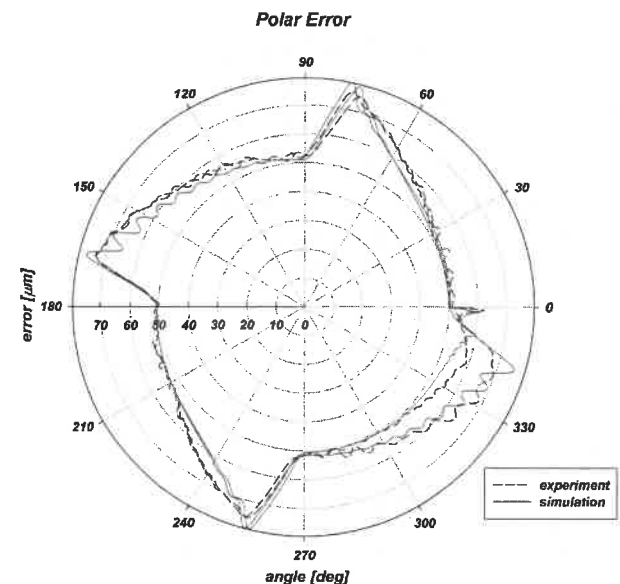


FIGURE 12. Circular interpolation response comparison of the XY stage.

CONCLUSIONS

We presented models and simulation results of multi axis machine tools. The complete multi-axis mechanical parts were modeled using blocks representing joints and constraints from SimMechanics and other parts are modeled by common Simulink block. Experiments were carried out for a 4-axis machine tool and for a commercial machine tool and for a XY stage. In all 3-case, simulation and experimental results showed good matches. The physical modeling approach for multi-axis machine tools is expandable to other configurations with ease of use.

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MODELING AND PARAMETER IDENTIFICATION OF GEOPHONE

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INSTRUCTIONS

Accurate absolute velocity sensing is critical for active vibration isolation (AVI) purposes [1]. Geophone is the most commonly used absolute velocity sensor in the field of AVI, where geophone resonance frequency should be as low as possible in order to measure low frequency vibration. Geophones with low resonance frequency always have large dimensions and need external power supply, which are not cost-effective. The resonance frequency of commercially available geophone that suitable for AVI application is typically at 4~12Hz. Therefore extension of geophone linear response range at low frequency is necessary, which is based on accurate model and parameters of geophone [2]. Manufacturers provide geophone parameters with an error interval, which makes parameter identification of geophone necessary. Shake-table method is one of the classic identification techniques, which needs a collocated, previously identified velocity sensor as reference sensor [3]. And it needs mechanical exciting equipment that limits its application in the field. Masahiro [4] and Frank [5] use impedance method to identify the parameters of a geophone. But geophone is a mechatronics system, thus injecting current can't excite its mechanical characteristics completely, which make impedance method less accurate than Shake-table method [3]. The step release method is a simple, fast and very accurate method for calibrating geophones when used with a system identifier [6]. But its precision is low when the geophone has high damping ratio [7].

In this paper both electromagnetic and mechanical principles of geophone are described and the geophone is modeled as a second order system. A new method for parameter identification of geophone is introduced. In the proposed parameter identification method the geophone to be identified and an accelerometer, which has linear frequency response down to DC, placed closely enough together that it can be assumed

that they record the same ground motion. With an accelerometer used as reference sensor the geophone frequency response to input acceleration can be measured, which is the key to the proposed parameter identification method. Geophone parameters can be identified by curve fitting.

GEOPHONE MODELING

Geophone is a highly sensitive electromagnetic velocity sensor. Generally, a geophone consists of a permanent magnet, a coil and a mechanical spring. There are two topologies of geophone, i.e. moving coil and moving magnet. This paper concerned with moving coil geophone. When the ground vibrates, the motion of the coil with respect to the magnet induces an EMF in the coil, which is proportional to the absolute velocity to be measured above the resonance frequency of the geophone.

Mechanical principles

As is shown in Figure 1, a geophone can be considered as a mass-spring-damper vibration system with single degree of freedom in its measurement axis.

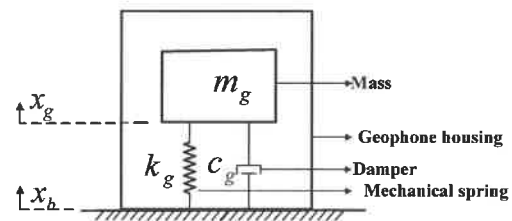


FIGURE 1: Geophone mechanical model

The equation of geophone mechanical part is given by

$$m_g \ddot{x}_g = -c_g (\dot{x}_g - \dot{x}_b) - k_g (x_g - x_b) + F \quad (1)$$

Where x_b and x_g are the vibration displacement of the ground and the coil respectively, F stands for Lorentz force generated by the induction current across the coil.

Electromagnetic principles

When the geophone housing accelerates with ground motion, the coil moves relative to the fixed permanent magnet that fixed to the

geophone housing, and induces a voltage across the coil, which is proportional to the relative velocity between the coil and the permanent magnet

$$U = G(\dot{x}_g - \dot{x}_b) \quad (2)$$

Where G is the constant of proportionality which is the generator constant that determined by geophone coil characteristics.

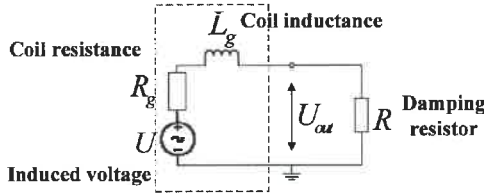


FIGURE 2: Geophone electrical model

If the geophone output leads are unconnected, no current flows and only mechanical damping exists. In practice the geophone would be parallel connected with an external resistor R , damping resistor, to add electrical damping. Then the equation of geophone electrical part, taking into account the voltage generated across the coil, is given by

$$U = i(R_g + R) - L_g \frac{di}{dt} \quad (3)$$

For a ground movement at frequency ω , the rate of the change of the current is $di/dt \approx \omega i$. The coil inductance will be negligible if $\omega L_g \ll (R_g + R)$. Usually the damping resistor is large enough for making the coil inductance negligible at the interested frequency band.

The current flows through the coil will generate a Lorentz force on it, which has the opposite direction with the coil motion.

$$F = -Gi \quad (4)$$

Transfer function

Combining both equations of the geophone mechanical and electrical part, the transfer function of the geophone output voltage to input vibration velocity is obtained as

$$G(s) = -G \frac{R}{R_g + R} \frac{m_g s^2}{m_g s^2 + (c_g + \frac{G^2}{R_g + R})s + k_g} \quad (5)$$

Rewrite the Eq. 5 as follows

$$G(s) = \frac{U_{out}(s)}{\dot{X}_b(s)} = \frac{-Gs^2}{s^2 + 2\zeta_g \omega_g s + \omega_g^2} \quad (6)$$

It can be seen from Eq.6 that geophone transfer function of output voltage to input velocity is a high pass filter, which is determined by the sensitivity constant G , the damping ratio ζ_g and the resonance frequency ω_g .

$$\omega_g = \sqrt{k_g / m_g} \quad (7)$$

Geophone manufacturers usually provide open-circuit parameters. With a damping resistor R connected to the output leads of geophone, the damping ratio and the sensitivity constant change.

$$G_R = G \frac{R}{R_g + R} \quad (8)$$

$$\zeta_R = \zeta_g + \frac{1}{2m_g \omega_g} \frac{G^2}{R_g + R} \quad (9)$$

TABLE 1: Parameters of SM6 4.5Hz geophone provided by manufacturer

Resonant frequency[Hz]	4.5±0.5
Open-circuit damping ratio	0.56± 5%
Proof mass[g]	11
Open-circuit velocity sensitivity [V/(m/s)]	28.8±5%
Coil resistance[Ω]	375±5%

The magnitude frequency response of output voltage to input velocity with and without damping resistor evaluated for a 4.5Hz SM6 geophone with the parameters described in Table 1.

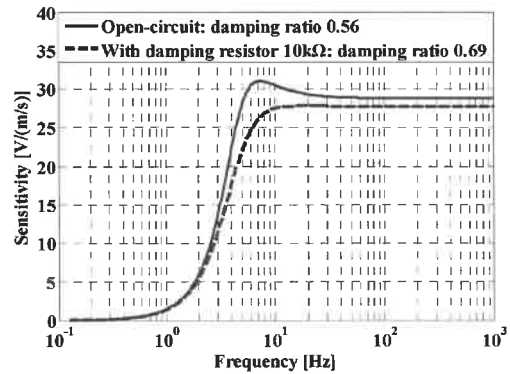


FIGURE 3: Geophone predicted magnitude frequency response to input velocity

Geophone transfer function of output voltage to input velocity is determined by the resonance frequency, the damping ratio and the sensitivity constant. Manufacturers provide these parameters with an error interval, which is

shown in Table 1. In applications like active vibration isolation, where velocity sensing is critical, the knowledge of geophone accurate parameters is essential. More importantly, linear frequency response range extension at low frequency of geophone is based on its accurate parameters.

GEOPHONE PARAMETER IDENTIFICATION

For a geophone whose resonance frequency is ω_g and damping ratio is ζ_g , a convenient method to extend its linear frequency response range at low frequency is to use a stretcher filter, with double poles at ω_c and double zeros at ω_g , where $\omega_c < \omega_g$. The filter transfer function is given by

$$C(s) = \frac{s^2 + 2\zeta_g\omega_g s + \omega_g^2}{s^2 + 2\zeta_c\omega_c s + \omega_c^2} \quad (10)$$

With the stretcher filter series connected to the geophone, the zeros of the filter cancel the poles of the geophone transfer function. Then the new transfer function of the geophone is obtained as

$$G_c(s) = G(s) \cdot C(s) = \frac{-Gs^2}{s^2 + 2\zeta_c\omega_c s + \omega_c^2} \quad (11)$$

The key to low frequency extension of geophone is the knowledge of its accurate parameters, based on which the zeros of the stretcher filter are designed. Assume the exact damping ratio of geophone is known, but the resonance frequency is known with an error interval such as 4.5 ± 0.5 Hz. When the stretcher filter is designed according to 4.5Hz regardless of the uncertainties of geophone resonance frequency, different extension results will be obtained, which are shown in Figure 4. Only when the parameters of geophone are matched well with that of the stretcher filter, it is possible to realize linear frequency response at low frequency. Therefore parameter identification of geophone is necessary.

The geophone is modeled as a second order system that model identification is unnecessary for geophone. As long as the input velocity signal is measured with a reference velocity sensor and the output voltage of the geophone is recorded simultaneously, geophone frequency response of output voltage to input velocity can be calculated. Then geophone parameters can be acquired by curve fitting. However, the parameters of the reference velocity sensor itself must be previously identified, which makes this kind of method based on measuring geophone

frequency response to input velocity inappropriate in many applications.

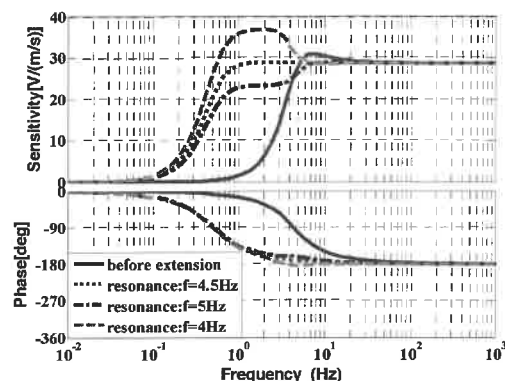


FIGURE 4: Results of linear frequency response extension of geophone at low frequency with a stretcher filter with double poles at $f=4.5$ Hz and double zeros at $f=0.5$ Hz

The transfer function of output voltage to input acceleration, which is given by Eq.12, is evaluated for geophone using the parameters described in Table 1.

$$G(s) = \frac{U_{out}(s)}{\ddot{X}_b(s)} = \frac{-Gs}{s^2 + 2\zeta_g\omega_g s + \omega_g^2} \quad (12)$$

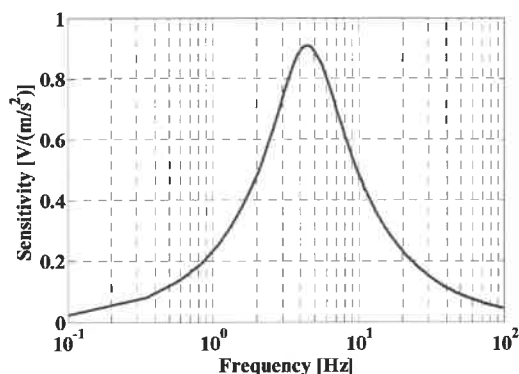


FIGURE 5: Geophone predicted magnitude frequency response to input acceleration

Eq.6 and Eq.12 noting that both transfer functions of geophone to input velocity and input acceleration are determined by the same parameters, which means that the geophone parameters can be obtained by measuring its response to input acceleration. Substituting $s=j\omega$ into Eq.12, it yields the magnitude and phase frequency response.

$$|G(j\omega)| = \frac{G}{\sqrt{\left(\frac{\omega_g^4}{\omega^2} + \omega^2 - 2\omega_g^2\right)^2 + 4\zeta_g^2\omega_g^2}} \quad (13)$$

$$\angle G(j\omega) = -90^\circ - \arctan \frac{2\zeta_g \frac{\omega}{\omega_g}}{1 - \left(\frac{\omega}{\omega_g}\right)^2} \quad (14)$$

Acquirement of geophone frequency response to input acceleration is the key to the proposed parameter identification method. Once it is measured, the resonance frequency, the damping ratio and the sensitivity constant can be obtained by curve fitting for its magnitude frequency response. The accuracy of the parameter identification can be validated according to the coincidence degree between the measured and fitted phase frequency response.

Generally, acceleration sensors have low pass characteristics, with excellent linearity, which are able to measure vibration acceleration from DC to thousands of Hertz theoretically. Therefore the input acceleration signal of the geophone can be measured with an accelerometer and the output voltage of the geophone should be recorded meanwhile.

Figure 6 shows the setup scheme to proposed parameter identification method. The geophone to be identified and the accelerometer should be placed closely enough together that it can be assumed that they record the same ground motion. Utilizing the input acceleration signal and the output voltage signal, geophone frequency response to input acceleration can be calculated by FFT. The model of the transfer function of output voltage to input acceleration is known, so parameter identification of geophone can be carried out by curve fitting.

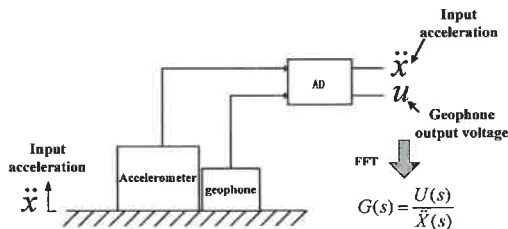


FIGURE 6: Setup scheme to proposed parameter identification method

Geophone resonance frequency, damping ratio and sensitivity constant are the object of the parameter identification. Repeating the

identification process with and without damping resistor R , whose value is known, geophone parameters can be identified in both circumstances. And utilizing the relationships given by Eq.8 and Eq.9, the coil mass and coil resistor value also can be obtained by simple calculation.

EXPERIMENTS

In proposed parameter identification method introduced in this paper, the input acceleration signal of accelerometer and geophone can be the ground vibration acceleration. Of course external excitation is recommended in order to fully excite the mechanical properties of the geophone, which also can improve the signal to noise ratio of the input signal. The geophone and the accelerometer placed closely enough so that they record the same ground motion. With PCB 393B31 accelerometer, whose sensitivity is $1.024 \text{ V/(m/s}^2\text{)}$, the input acceleration signal is measured. Data acquisition is fulfilled by a 24bit AD device. Its resolution is $1.920928 \times 10^{-6} \text{ V}$ for $\pm 10 \text{ V}$ range. Utilizing the geophone output voltage and the accelerometer output acceleration signal, geophone frequency response of output voltage to input acceleration is calculated by FFT with a hundred times linear average with 50% overlap.

Results of parameter identification experiment and curve fitting with and without damping resistor are shown in Figure 7 and Figure 8 respectively. The geophone parameters are obtained by curve fitting. The identified parameters with and without damping resistor are given in Table2.

TABLE 2: Results of parameter identification with and without damping resistor

	Open-circuit	with damping resistor $R=10k \Omega$
Resonant frequency[Hz]	4.876	4.875
damping ratio	0.5611	0.6867
Sensitivity constant [V/(m/s)]	30.02	28.88

The coincidence degree between the measured and fitted phase frequency response can be used to verify the accuracy of the identification. It can be seen from Figure7 and Figure8 that the

identified parameters describe the geophone's characteristics very well.

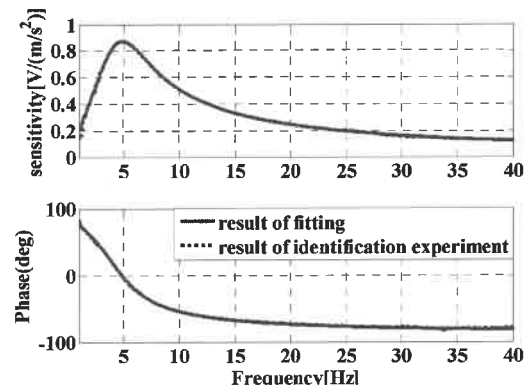


FIGURE 7: Open circuit geophone frequency response to input acceleration. Dotted line: measured frequency response using proposed identification method. Solid line: result of fitting

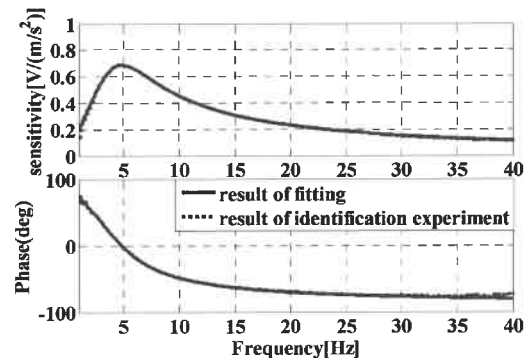


FIGURE 8: Frequency response to input acceleration of geophone with damping resistor $R=10k\ \Omega$. Dotted line: measured frequency response using proposed identification method. Solid line: result of fitting

CONCLUSION

Geophone linear frequency response range extension at low frequency is based on accurate model and parameters of the geophone. In this paper both electromagnetic and mechanical principles of the geophone are introduced. The geophone is modeled as a second order system. In order to obtain the exact parameters of geophone, a new parameter identification method is established. In proposed method, geophone frequency response to input acceleration is measured. And the parameters of geophone are identified by curve fitting. The effectiveness of proposed identification method is validated by identification experiments.

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COMBINED NONLINEAR FEEDBACK AND CASCADED ITERATIVE LEARNING CONTROL FOR AN ULTRA-PRECISION DUAL-STAGE ACTUATED WAFER STAGE

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INTRODUCTION

This paper presents a coordinated feedback and feedforward control design approach for the developed wafer stage, which employs a dual-stroke principle with a short-stroke stage for fine positioning and a long-stroke stage for coarse positioning [1]. Given the repetitive nature of wafer stage, iterative learning control (ILC) is utilized to realize precision motion control.

ILC refers to the iterative process of obtaining the optimal feedforward signal under repetitive motion. Previous information such as tracking error and learned feedforward is incorporated to form the feedforward signal of the next execution of motion [2]. However, as a feedforward technique in time domain, ILC has the character of amplifying the nonrecurring disturbances during the process of attenuating the recurring. Therefore, for improved the performance of ILC, it is essential to decrease the nonrecurring contents in the tracking error and improve its ability to eliminate the recurring disturbances simultaneously.

In terms of the aspect of decreasing the nonrecurring contents in the tracking error, as an alternative [3],[4], we propose the use of nonlinear feedback controller instead of a linear one. Since nonlinear feedback controller will not subject to the Bode's sensitivity integral theorem. So low-frequency disturbances attenuation can be combined with high-frequency noise suppression, enhancing the reproducibility of the tracking error both in low-frequency and high-frequency.

While in terms of the aspect of attenuating recurring disturbances, an cascaded learning scheme is developed to further decrease the residual error resulting from a limited bandwidth of Q-filter. Once the tracking error converges to

the predefined threshold, the learned feedforward signal is frozen and injected to the control loop as a constant feedforward force. Based on the updated initial effort, a new learning process begins until no more improvement can be achieved.

To summarize, the nonlinear feedback controller guarantees the system stability and addresses the nonrecurring, low-frequency disturbances in the time domain, while the cascaded ILC compensates for the recurring disturbances in the iteration domain. And the proposed control scheme is demonstrated on the developed wafer stage with the settling time decrease from 100ms to 4ms.

NONLINEAR FEEDBACK CONTROL BASED CASCADED ILC

Due to its merit of not requiring a accurate plant model yet still yielding high tracking performance, ILC has been widely employed in the occasions requiring repetitive works such as industrial robots, semiconductor fabrication, and LED manufacturing. As a feedforward technique in time domain, ILC has the character of attenuating the recurring disturbances such as trajectory input, cable disturbances and force ripple but at the cost of amplifying the nonrecurring such as measurement noise and stochastic external disturbances. Therefore, it is usually utilized in combination of feedback control to address the stochastic disturbances and stabilize the system.

Aimed at improved convergence accuracy of ILC, in this paper a nonlinear feedback control based cascaded ILC scheme is proposed, as depicted in Fig.1. On the one hand, nonlinear feedback control is utilized to improve the reproducibility of tracking error, alleviating the side effect of ILC in amplifying nonrecurring disturbances. On the

other hand, an cascaded learning scheme is developed to further decrease the residual error, improving the ability of ILC in eliminating the recurring disturbances.

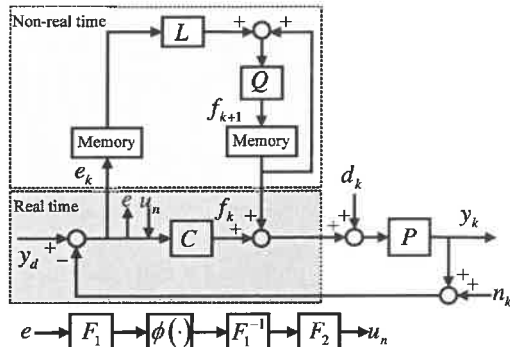


FIGURE 1. The nonlinear feedback based ILC control scheme.

Nonlinear Feedback Control

For many motion systems, linear feedback controller are used. However, the linear feedback control is subjected a fixed trade-off between low-frequency disturbances attenuation and high-frequency noise amplification. One of the alternatives is to utilize the nonlinear feedback controller for the linear system, see Heertjes [5]. In this paper, nonlinear controller is incorporated to improve the reproducibility of the tracking error, which is the premise of the implementation of ILC.

In the Fig.1, C and P denotes the linear feedback controller and plant, respectively. In the feedback control loop, a nonlinear controller is added before the nominal linear feedback controller C with the expression given by

$$N = (1 + F_2 F_1^{-1} \phi(\cdot) F_1) \quad (1)$$

$$\phi(\cdot) = \begin{cases} a(1 - \frac{\delta}{|e|}) & , |e| > \delta \\ 0 & , |e| \leq \delta \end{cases} \quad (2)$$

with e denotes the tracking error, u_n the output of the nonlinear controller. F_1 is a loop shaping filter to select the interested frequency range to be attenuated, while F_2 is used to eliminate the possible high-frequency noise resulting from the gain mutation.

Within a frequency range of interest, the gain is continuously updated to suppress the external disturbances. Based on the frequency contents of error signal, for error-levels beyond the

threshold δ , additional learning gain will be effectively used to diminish the low-frequency error. Below the threshold, the original low-gain value is maintained to avoid high-frequency noise amplification.

The parameter α is tuned through data-based optimization methods [6]. While the threshold δ and the bandwidth of the filter F_1 , which determines the frequency range of nonrecurring disturbances, are identified based on the tracking error reproducibility analysis. Under the linear feedback controller C , the tracking error e can be written as

$$e = S y_d - S n - S_p d \quad (3)$$

where y_d is the trajectory input which is trial-invariant, n are stochastic noise and d are external disturbances which consists of recurring disturbances d_r and nonrecurring d_{nr} . Based on the assumption that the nonrecurring disturbances are zero means, the recurring part of tracking error e^r can be approximated by averaging the tracking error of h sequential trials.

$$e^r = S y_d - S_p d_r \approx \sum_{j=1}^h e_j \quad (4)$$

where S is the sensitivity function and S_p the process sensitivity function. Then it yields the corresponding nonrecurring part of tracking error of each trial as

$$e_j^{nr} = e_j - e^r \quad (5)$$

With the nonrecurring error of each trial, the energy distribution in frequency domain can be examined by means of cumulative spectrum square.

Cascaded ILC

For the feedforward control, ILC is utilized with the tracking error e_k and feedforward signal f_k stored and then updated by the learning filter L , obtaining the feedforward signal of next trial f_{k+1} .

$$f_{k+1} = Q(f_k + L e_k) \quad (6)$$

And

$$e_k = \frac{1}{1 + NCP} (y_d - n_k - P d_k) - \frac{P}{1 + NCP} f_k \quad (7)$$

$$= \hat{S} r_k - \hat{S}_p f_k$$

Then it yields the tracking error of next trial

$$e_{k+1} = Q(1 - \hat{S}_p L) e_k + \hat{S} (Q r_{k+1} - r_k) \quad (8)$$

When k tends to be infinite, the asymptotic error is deduced as

$$e_{\infty} = \frac{(r_{k+1} - Qr_k)}{1 - Q(1 - \hat{S}_p L)} \hat{S} \quad (9)$$

As the trajectory input dominates over other disturbances, and assume that L approximate the reverse of process sensitivity in low-frequency, the asymptotic error can be simplified as

$$e_{\infty} \approx (1 - Q) \hat{S} y_d \quad (10)$$

From Eq. 10 it can be seen that the Q-filter determines the attainable convergence accuracy. Only if the value of Q-filter is equal to one, the tracking error can be completely eliminated. However, in order to guarantee the robustness against higher order dynamics and suppress the noise amplification, Q usually is a zero-phase low-pass filter with a limited bandwidth. Fig. 2 shows its bode magnitude characteristics.

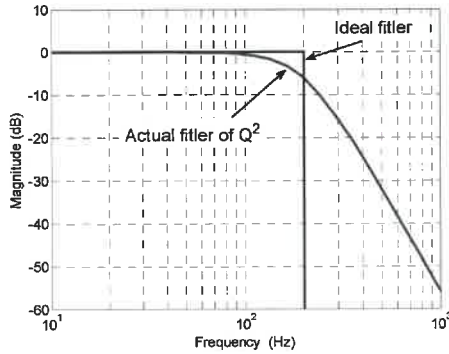


Figure 2. Magnitude characteristics of the zero-phase low-pass Q-filter with a cut-off frequency of 200Hz.

The bode magnitude representation shows two important properties. Firstly, the magnitude of the zero-phase Q-filter is close to zero in high-frequency, which means that the high-frequency contents of e_0 will not be eliminated. Secondly, the magnitude under the cut-off frequency is not entirely equal to one. As the practical trajectory contains more energy in low frequencies, this part of tracking error will contribute more to the residual error.

To obtain robustness without sacrificing the tracking performance, a cascaded learning scheme is proposed. The key of the method is the feedforward frozen mechanism, as illustrated in Fig. 3. The update law is modified as

$$u_{k+1} = f_{k+1} + \Gamma_m \quad (11)$$

where u_{k+1} is the learned feedforward signal for the next trial, Γ_m denotes the previous learned feedforward signal with m as the index number. Once the tracking error attenuation is lower than the predefined threshold η , the learned feedforward signal f_k is frozen and added to Γ_m , which is injected to the control loop as a constant feedforward signal. Meanwhile, f_k is emptied to zero.

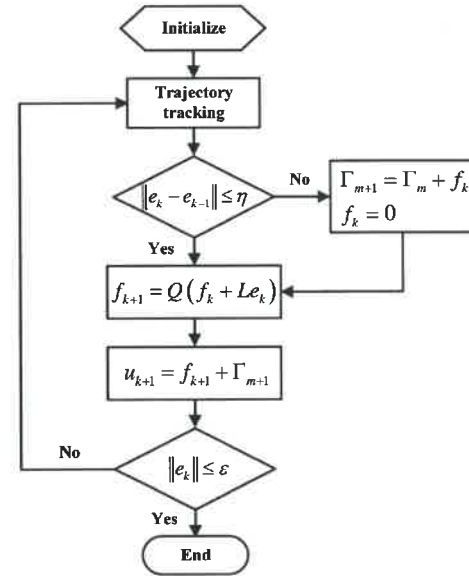


Figure 3. The illustration of cascaded ILC

Assume after M trials, the learning converges to its limit, the tracking error becomes

$$e_M = (1 - Q) \hat{S} y_d \quad (12)$$

Then the learned feedforward signal is frozen and injected to the control loop as a constant feedforward. Based on the frozen feedforward signal, new iterative learning process begins with the initial error of e_M . Therefore, after M trials the tracking error will not decrease again, and the residual error will evolve as

$$e_{2M} = (1 - Q) e_M = (1 - Q)^2 \hat{S} y_d \quad (13)$$

The same operations are repeated until no more improvement can be obtained. Therefore, after the learning process repeated m times, the residual error will be

$$e_{nM} = (1 - Q)^{n-1} e_{(n-1)M} = (1 - Q)^n \hat{S} y_d \quad (14)$$

when n tends to be infinite, it yields to the asymptotic error

$$e_{\infty} = \lim_{n \rightarrow \infty} e_{nM} = \lim_{n \rightarrow \infty} (1 - Q)^n \hat{S} y_d = 0, \quad Q \leq 1 \quad (15)$$

It can be concluded that the tracking error resulting from the trajectory input can be completely eliminated by employing cascaded iterative learning control scheme.

EXPERIMENTS

In this section, the performance of proposed control scheme is assessed on our developed wafer stage. The goal is to achieve constant velocity as quickly as possible to reach a specified accuracy so that scanning process can be performed.

The Developed Wafer Stage

The developed wafer stage includes a short-stroke stage for fine positioning and a long-stroke stage for coarse positioning. Fig. 4 depicts the exploded view of a computer-aided design (CAD) model of the developed wafer stage, and Fig. 5 displays the actual assembly of the wafer stage.

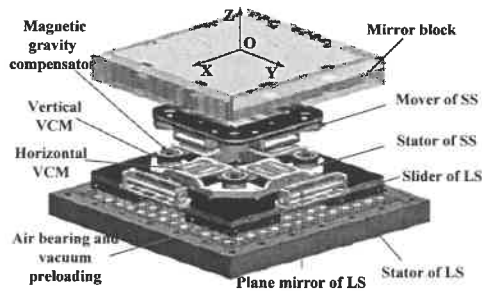


FIGURE 4. The schematic of the designed wafer stage



FIGURE 5. The photograph of developed wafer stage

The short-stroke stage is a magnetic levitated stage driven by eight voice coil motors (VCM). The characteristics, such as light weight and no thrust force, allows for higher bandwidth and 6-DOF fine positioning. The long-stroke stage is a planar actuator which moves along the surface of a Halbach permanent magnet array without generating friction due to being elevated with air

bearings. By stacking the short-stroke stage on the long-stroke stage, nano-positioning accuracy is combined with large travel range.

Trajectory Planning

To begin with, fourth order trajectory is designed which can be seen in Fig. 6. The trajectory has a cruising speed of 250mm/s and a maximum acceleration/deceleration of 10 m/s^2 . The servo period is $200\mu\text{s}$, and the resolution of the laser interferometer is 0.6nm.

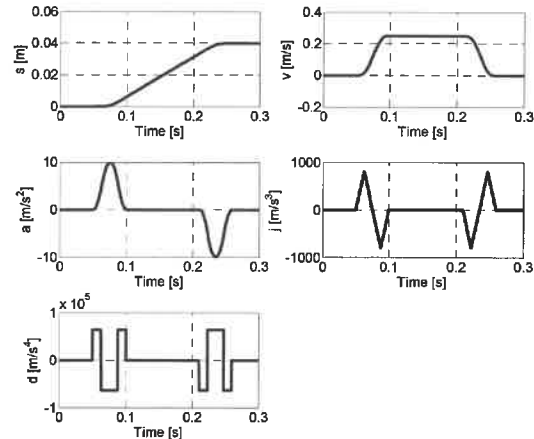


Figure 6. The planned fourth order trajectory

Implementation of Standard ILC

In this subsection, standard ILC is implemented for the wafer stage. Fig. 7 shows the convergence curve of the tracking error. After 6 trials, the tracking error converges to its limit and the learned feedforward signal stops increasing.

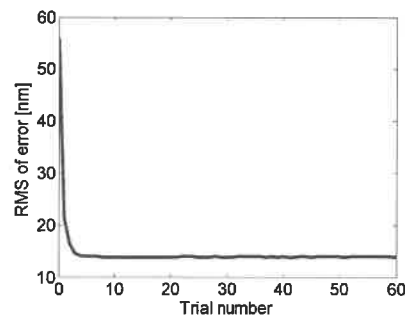


Figure 7. Tracking error under standard ILC in scanning direction

Then, the reproducibility of the residual tracking error is examined by repeating 10 trials with the same trajectory and learned feedforward, as shown in Fig. 8. The upper plot is the recurring error by averaging the tracking error of 10 trials, while the lower is the nonrecurring error. It can

be found that after 10 trials, the amplitude of the recurring error is still larger than the one of nonrecurring, leading to the conclusion that the recurring error still dominates over the nonrecurring and the tracking error is not sufficiently eliminated by standard ILC.

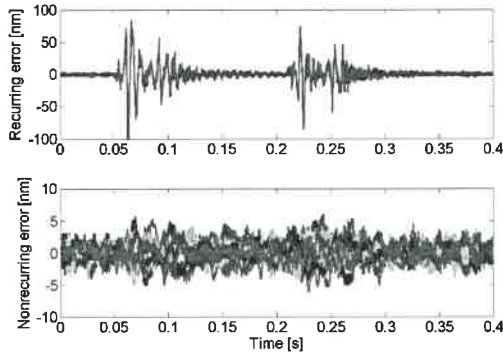


Figure 8. The recurring (upper plot) and nonrecurring (lower plot) tracking error after 10 trials under standard ILC

Tracking Performance Comparison

To further improve the tracking performance, the cascaded ILC is applied. Figure 9 shows the tracking error of 30th trial. Table 1 presents the detailed tracking performance, demonstrating its effectiveness in reducing settling time and eliminating tracking errors.

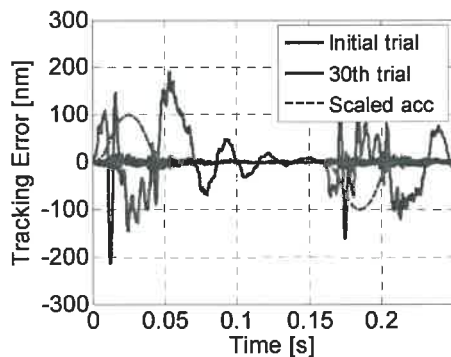


FIGURE 9. The comparison of tracking error between initial trial and 30th trial

TABLE 1. The tracking performance comparison between initial trial and 30th trial

Performance indicators	Initial trial	30 th trial	Units
Max. tracking error	210	22	[nm]
Settling time	100	4	[ms]

CONCLUSIONS

Aimed at improved convergence accuracy of ILC, a nonlinear feedback control based cascaded learning scheme is put forward. Specifically, the nonlinear feedback control balances the trade-off between low-frequency disturbances attenuation and high-frequency noise amplification, improving the reproducibility of the tracking error. Furthermore, the cascaded learning scheme is proposed to improve the ability of ILC to diminish the recurring disturbances. The experiments on the developed wafer stage demonstrate that the proposed algorithm can remarkably improve the tracking performance with the settling time reduced from 100ms to 4ms.

ACKNOWLEDGEMENT

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DESIGN OF AN INVERSE MODEL FEEDFORWARD CONTROLLER BASED ON IDENTIFICATION MODELS OF A VIBRATION ISOLATOR

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INTRODUCTION

Low frequency vibration can affect the accuracy, repeatability and throughput of precision equipment. Various active vibration isolation control strategies, such as PID control, pole assignment, fuzzy control, and sliding control have been proposed and have already been used in many applications [1-4]. Moreover, feedforward control strategy has been widely used in active vibration isolation control and has achieved good performance [5]. The most-used Fx-LMS feedforward strategy ensures stability of the system, while the dynamic performance of the system can only be approximated by zeroes. However, for a system with a complex dynamic performance, the Fx-LMS feedforward strategy sometimes does not converge [6]. Also, Fx-LMS feedforward strategy is efficient if the input signal of a system changes significantly, but is not as good as desired if the input signal is steady. An inverse model feedforward controller is obtained by models which include poles and can approximate the dynamic response of a system more accurately. Moreover, it can be easily realized by circuit or computer programs. However, the ideal inverse model feedforward controller cannot be applied in active vibration isolation directly, because it is not always stable. Aimed at improving the active vibration isolation performance, a new inverse model feedforward controller design based on identification models is proposed in this paper. To design a stable and realizable feedforward controller, compensation and parameter design method are discussed. The effectiveness of the feedforward control strategy is demonstrated by simulation and experimental results and is compared with absolute velocity feedback.

PRINCIPLE OF ACTIVE VIBRATION CONTROL

Consider a vibration isolation system as shown in Figure 1.

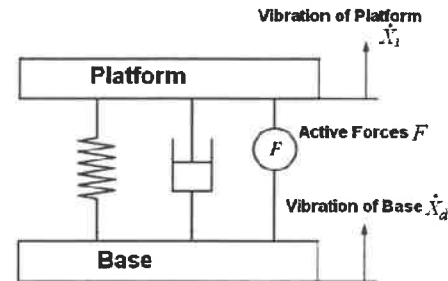


Figure 1. Basic model of a vibration isolation system

The system is determined by the primary path:

$$G_d(s) = \frac{\dot{X}_i(s)}{\dot{X}_d(s)}$$

and the secondary path:

$$G_F(s) = \frac{\dot{X}_i(s)}{F(s)}$$

Assuming the system is linear and two transfer paths are independent, the vibration of platform can be expressed as:

$$\dot{X}_i(s) = G_d(s)\dot{X}_d(s) + G_F(s)F(s)$$

Using active vibration control, the actuators generate vibration control forces F . The vibration $G_F(s)F(s)$ generated by the primary path and the vibration $G_d(s)\dot{X}_d(s)$ generated by secondary path counteract with each other to achieve vibration reduction.

Absolute velocity feedback control strategy can eliminate the resonance peak by increasing system damping and can reach an attenuation of -20dB at 10Hz [7]. Feedforward control is more efficient in compensating external disturbances. When disturbances happen, the active forces generated by feedforward controller can control the system before the disturbances affect it. It can improve the vibration isolation performance to meet the

micro-nano manufacturing and precision equipment requirements.

Figure 2 shows an active control scheme with feedback and feedforward strategy.

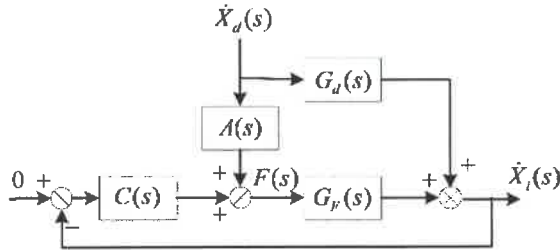


Figure 2. Active control scheme with feedback and feedforward strategy

The transmissibility $I(s)$ is used to evaluate the performance of a vibration isolation system in general. $I(s)$ is defined as:

$$I(s) = \frac{\dot{X}_i(s)}{\dot{X}_d(s)} = [I + G_F(s)C(s)]^{-1} [G_d(s) + G_F(s)A(s)]$$

$$= [I + G_F(s)C(s)]^{-1} G_d(s) [I + G_d(s)^{-1} G_F(s)A(s)]$$

$$= G_{FB}(s) G_{FF}(s)$$

In the expression above, the feedback function $G_{FB}(s)$ and the feedforward function $G_{FF}(s)$ are independent since they contain the feedback controller $C(s)$ and the feedforward controller $A(s)$ respectively. On this account, $C(s)$ and $A(s)$ can be designed separately, namely, a feedforward controller can be added to an active vibration system $G_{FB}(s)$ with a feedback controller which is already designed. In order to improve the performance of a vibration isolation system, a feedforward controller design based on identification model is proposed.

DESIGN OF INVERSE MODEL FEEDFORWARD CONTROLLER

An inverse model controller is obtained by models which include poles in comparison with Fx-LMS strategy. It can approximate the dynamic response of a system more accurately. For an ideal vibration isolation system, the vibration of the platform is zero. Therefore, the ideal inverse model feedforward controller can be obtained by $G_F(s)$ and $G_d(s)$ as:

$$A(s) = -G_F(s)^{-1} G_d(s)$$

When designing the controller, identification models or estimation models of $G_F(s)$, $G_d(s)$ are used. However, the ideal inverse model feedforward controller cannot be applied directly

mainly because $A(s)$ is not always stable. In other words, $A(s)$ has poles in the right-half of the s-plane. To solve the problem, a method to modify $A(s)$ is proposed to ensure stability of the controller.

The ideal inverse model feedforward controller can be written as:

$$A(s) = \frac{A_n(s)}{A_{d-}(s)A_{d+}(s)}$$

$A_n(s)$ is the numerator polynomial.

$$A_{d-}(s) = K_d(s-p_1)(s-p_2) \cdots (s-p_k)$$

is polynomial of poles in the left-half of the s-plane, and

$$A_{d+}(s) = (s+p_{k+1}) \cdots (s+p_l)$$

is polynomial of poles in the right-half of the s-plane. The form of the modified feedforward controller $A'(s)$ is proposed as:

$$A'(s) = \frac{A_n(s)}{A_{d-}(s)} B(s)$$

The controller $A'(s)$ will also be stable if the function $B(s)$ is stable. $B(s)$ compensates the poles to ensure that the denominator order is larger than the numerator order of the controller [8]. Moreover, different forms and parameters of $B(s)$ lead to different vibration isolation performance. The optimal parameters design for $B(s)$ is necessary for performance improvement and is given in the next section.

PARAMETER DESIGN OF COMPENSATION FUNCTION

Substituting $A'(s)$ into $G_{FF}(s)$ gives the expression for the feedforward function:

$$G_{FF}(s) = 1 - A_{d+}(s)B(s)$$

To determine parameters of $B(s)$, an optimization function in the frequency range (ω_0, ω_1) is proposed as:

$$g[B^*(s)] = (g[B(s)])_{\min} = \left(\int_{\omega_0}^{\omega_1} |G_{FF}(j\omega)|^2 d\omega \right)_{\min}$$

The function $g[B(s)]$ describes the isolator's performance in (ω_0, ω_1) . $g[B(s)]$ achieves the minimum value when $B(s)$ gets the optimization result $B^*(s)$.

Consider a system which has poles in the right-half of the s-plane:

$$A_{d+}(s) = (s+p_{k+1}) \cdots (s+p_l)$$

$B(s)$ is proposed as:

$$B(s) = \frac{1}{\left(\frac{1}{\tau}s + 1\right)^r} \frac{1}{K(s - p'_{k+1}) \cdots (s - p'_l)}$$

The low pass filter in $B(s)$ is:

$$LF(s) = \frac{1}{\left(\frac{1}{\tau}s + 1\right)^r}$$

It compensates the poles to ensure that the denominator order is larger than the numerator, in which the cutoff frequency $\tau=5\omega\sim 10\omega$ is chosen.

$$r = \deg(A_n(s)) - \deg(A_{d+}(s)A_{d-}(s))$$

is the difference between the denominator order and the numerator order of $A(s)$. Parameters $K, p'_{k+1}, \dots, p'_l$ are determined by the optimization function.

Consider a system which has a real pole $-p_0$ in the right-half of the s-plane:

$$A_{d+}(s) = s + p_0, p_0 < 0$$

Then

$$B(s) = \frac{1}{K(s - a)} \frac{1}{\left(\frac{1}{\tau}s + 1\right)^r}, a \leq 0.$$

In the frequency range of interest $1\sim 100$ Hz or the circular frequency range $2\pi\sim 200\pi$ rad/s, the optimization function can be written as:

$$g(K, a) = \int_{2\pi}^{200\pi} \frac{(K-1)^2 \omega^2 + (p_0 + Ka)^2}{K^2(\omega^2 + a^2)} d\omega, a \leq 0$$

If we only consider two cases, $K-1=0$ and $p_0+Ka=0$, $G_{FF}(s)$ and $B(s)$ can be obtained according to the optimization function:

$$B(s) = \frac{1}{s} \frac{1}{\left(\frac{1}{\tau}s + 1\right)^r}, G_{FF}(s) = 1 - \frac{s + p_0}{s} \frac{1}{\left(\frac{1}{\tau}s + 1\right)^r}, \text{ if } |p_0| < 20\pi$$

$$B(s) = \frac{1}{p_0} \frac{1}{\left(\frac{1}{\tau}s + 1\right)^r}, G_{FF}(s) = 1 - \frac{s + p_0}{p_0} \frac{1}{\left(\frac{1}{\tau}s + 1\right)^r}, \text{ if } |p_0| > 116\pi$$

Similarly, when the poles in the right-half of the s-plane are conjugate poles, $A_{d+}(s) = s^2 - 2\zeta\omega_0 s + \omega_0^2$. $G_{FF}(s)$ and $B(s)$ are determined as:

$$B(s) = \frac{1}{s^2} \frac{1}{\left(\frac{1}{\tau}s + 1\right)^r}, G_{FF}(s) = 1 - \frac{s^2 - 2\zeta\omega_0 s + \omega_0^2}{s^2} \frac{1}{\left(\frac{1}{\tau}s + 1\right)^r}, \text{ if } \omega_0 < 20\pi$$

$$B(s) = \frac{1}{\omega_0^2} \frac{1}{\left(\frac{1}{\tau}s + 1\right)^r}, G_{FF}(s) = 1 - \frac{s^2 - 2\zeta\omega_0 s + \omega_0^2}{\omega_0^2} \frac{1}{\left(\frac{1}{\tau}s + 1\right)^r}, \text{ if } \omega_0 > 20\pi$$

Also, a system with more than two poles in the right-half of the s-plane:

$$A_{d+}(s) = (s + p_{k+1}) \cdots (s + p_l)$$

The compensation function $B(s)$ is determined as:

$$B(s) = \frac{1}{s^{t-k}} \frac{1}{p_{t+1} \cdots p_l} \frac{1}{\left(\frac{1}{\tau}s + 1\right)^r},$$

$$|p_{k+1}| < \cdots < |p_t| < 20\pi < |p_{t+1}| < \cdots < |p_l|$$

At this point, the feedforward function $G_{FF}(s)$ is expressed as:

$$G_{FF}(s) = 1 - A_{d+}(s)B(s) \\ = 1 - \frac{(s + p_{k+1}) \cdots (s + p_t)(s + p_{t+1}) \cdots (s + p_l)}{s^{t-k} p_{t+1} \cdots p_l} \frac{1}{\left(\frac{1}{\tau}s + 1\right)^r}$$

SIMULATION OF INVERSE MODEL FEEDFORWARD CONTROLLER

Consider a vibration isolation system which has the identification model of primary path as:

$$G'_d(s) = \frac{-0.0017032(s + 93.52)(s^2 + 14.16s + 97.56)}{(s^2 + 5.04s + 131.3)(s^2 + 5.175s + 616.3)}$$

The identification model of secondary path:

$$G'_r(s) = \frac{-0.00036613(s - 3.078)(s^2 + 4.894s + 674.7)(s^2 - 2996s + 3597000)}{(s^2 + 6.659s + 542.5)(s^2 + 13.18s + 1000)(s^2 + 535.9s + 140200)}$$

According to the proposed design method, $B(s)$ is obtained as:

$$B(s) = \frac{1}{3597000s \left(\frac{1}{1000\pi}s + 1 \right)^2} = \frac{1}{0.3645s(s + 3142)^2}$$

In order to verify the efficiency of the proposed feedforward control strategy, performances by feedback control with and without feedforward control strategy are compared. The velocity transmissibility curves and the platform velocity response are demonstrated in Figure 3 and 4 respectively.

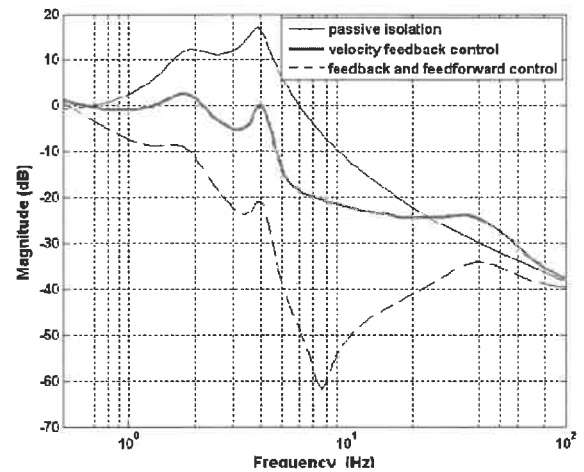


Figure 3. Velocity transmissibility for different control strategies

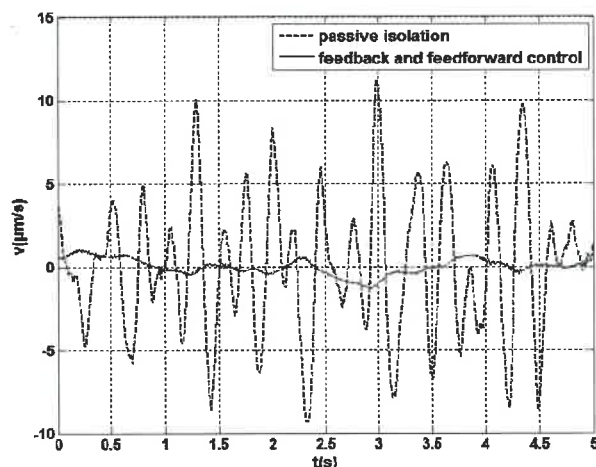


Figure 4. Velocity response at the center of the platform for passive control and feedback with feedforward control strategies

The simulation result in Figure 3 shows that the feedback control with feedforward strategy outperforms the feedback control alone at middle frequency which achieves a reduction of 40dB at 8Hz. However, both control strategies perform worse at ultralow frequency and the transmissibility tends to be the same at high frequency. The simulation result of velocity response is shown in Figure 4. Compared to the maximum velocity of 10 μ m/s at passive isolation mode, the low-frequency components are eliminated, and the vibration is reduced to $\pm 1.2\mu$ m/s, or about 15% at maximum in the feedback and feedforward control mode. The simulation results verify that the proposed feedforward control strategy is effective in performance improvement of vibration isolation.

EXPERIMENT WITH INVERSE MODEL FEEDFORWARD CONTROLLER

Figure 5 shows the control and measurement system of the vibration isolator. The 1m $\times$ 70cm vibration isolation platform supported on four air springs is the object of control. Active control forces are provided to the platforms through four actuators according to the velocity signals from the sensors. The performance of the isolator is measured by a dynamic signal analyzer with two acceleration sensors and an exciter. Random excitation force is generated by the exciter during the measuring and it is regarded as the vibration source to increase the signal to noise ratio of the measurement result.

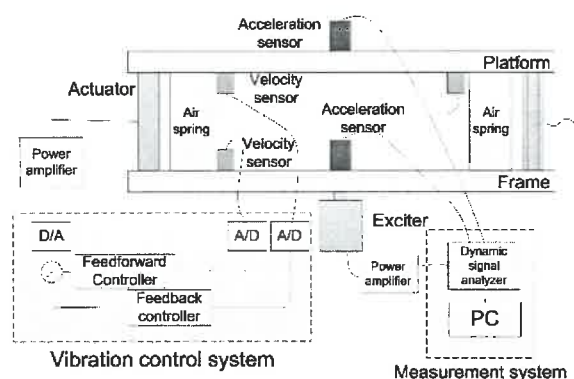


Figure 5. Control and measurement system of the vibration isolator

The system contains four velocity sensors measuring vibration of platform, one velocity sensor measuring vibration of frame and four actuators. A 4 by 1 model matrix of primary path $G_d(s)$ and a 4 by 4 model matrix of secondary path $G_f(s)$ were identified. The feedforward controller $A'(s)$ was designed by the method proposed above. The control frequency was 1 kHz and the performances for feedback control with and without feedforward strategy were measured in real time.

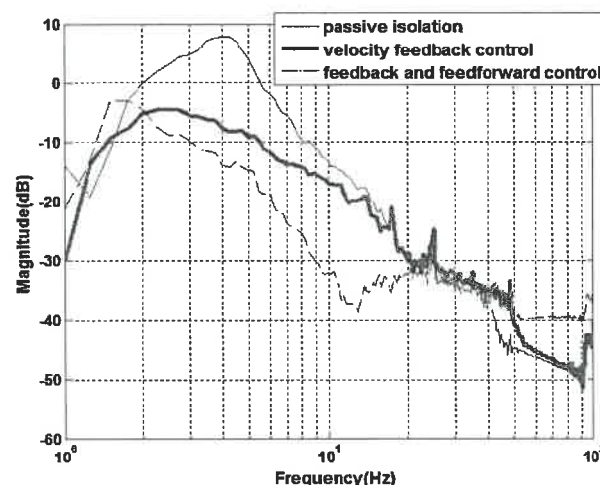


Figure 6. Velocity transmissibility for different control strategies

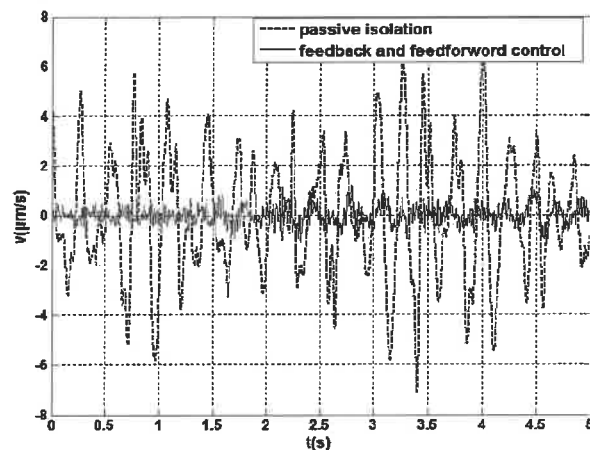


Figure 7. Velocity response at the center of the platform for passive isolation and feedback with feedforward control strategies

Figure 6 shows the velocity transmissibility from the frame to the vertical velocity response at the center of the platform for two different control strategies. When the system was controlled by absolute velocity feedback alone, vibration in the vicinity of the resonance frequency was attenuated below 0dB and the efficiency of isolation started at frequencies as low as 2 Hz. When feedforward control was applied, the isolation performance improved by -5 to -20 dB over 2 Hz and attained -30 to -40 dB at frequency over 9 Hz. Figure 7 shows the velocity response at the center of the platform in comparison with feedforward control and passive isolation. The proposed control strategy is very effective for vibration isolation as the vibration reduction reached 1/10 at maximum.

CONCLUSIONS

This paper discussed an inverse model feedforward controller design method based on identification model of vibration isolation system. To design a stable and realizable feedforward controller, the poles in the right-half of the s-plane of the controller were omitted and a compensation function was added. Moreover, parameters of the compensation function were determined by an optimization function which describes the performance of the vibration isolator. Through simulations and experiments, the efficiency of the feedforward controller was verified. When the vibration isolation system was controlled under feedback and feedforward strategy, an attenuation of -30 to -40 dB was achieved at frequency over 9Hz, and the efficiency of isolation starts at frequency as low as 2 Hz, or even less. The low-frequency

vibration enhanced in passive isolation mode was eliminated and the maximum velocity was reduced 90%.

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Design of a Mechanical Actuator for Phase Control of an X-ray Interferometer

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INTRODUCTION

In 1965, Bonse and Hart introduced the idea of phase-contrast micrography in their paper on the uses of an x-ray interferometer as a new imaging instrument [1]. Adding the phase-contrast process to traditional absorption imaging creates the ability to quantitatively measure x-ray phase shifts that will indicate composition of the medium that the x-rays pass through. Phase-contrast data has the potential to enhance the resolution of tradition absorption x-ray images, particularly for small specimens or when imaging weakly absorbing materials.

Primarily being developed for biological applications, the construction of phase-contrast enabled x-ray interferometers are currently restricted to high energy beam systems (synchrotrons) and have not been implemented for local laboratory measurements. The goal of this research project is to create a local laboratory-based system so that precise and robust measurements can be obtained for potential biological, micro-manufacturing, and forensic applications.

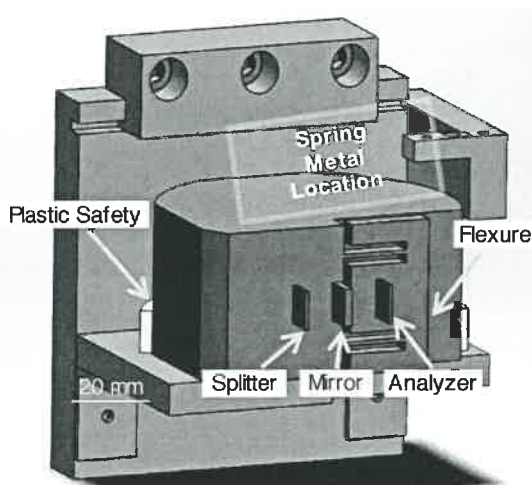


FIGURE 1. Silicon x-ray interferometer monolith (spring metal not shown).

A standard phase control mechanism for use in an x-ray interferometer and micro-CT systems is required for repeatable and accurate measurement of both phase-contrast and absorption-based images. Ultimately, the system designed within the scope of this project will be used for minimally destructive 3-dimensional imaging of sub-millimeter scale structures. The system will also be adaptable to accept x-ray interferometer monoliths manufactured from a variety of materials (i.e. silicon, YAG, or other garnet-based crystals) and geometries. This versatility will enable additional projects to utilize the same subsystem design for implementation of instruments with expanded scope. A solid model of the frame for supporting the single crystal silicon interferometer is shown in Figure 1.

THEORY

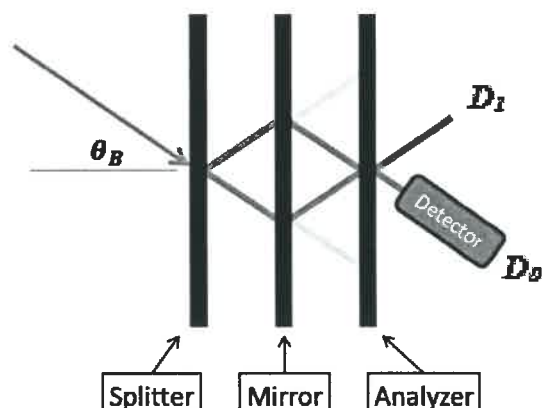


FIGURE 2. Schematic of the interferometer showing splitting and mirroring of x-ray beam.

In operation, an object is placed in one of the beam paths between the mirror and analyzer blades. The analyzer blade flexure is then translated. The split x-ray beams (shown green in Figure 2) will converge together at the analyzer blade that modulates outputs to the two detectors. If the object is thicker at one point, the

x-ray waves will experience a larger absorption resulting in weak D_0 and D_1 outputs as labeled in Figure 2. If a thinner section of the object is scanned, the x-ray waves will generate a wavefield of weak absorption and refracting large D_0 and D_1 outputs [1].

In this implementation, imaging of specimen will be achieved by scanning a specimen in one of the sub-millimeter diameter beams in the interferometer followed by 'tiling' these measurements to obtain image information over larger areas. For each sub-image, phase will be measured by scanning the interferometer and extracting relative phase between that of the crystal lattice and the specimen. This is necessary because specimens will have differing thicknesses and absorption characteristics. The phase of the interferometer can be obtained by translating a single crystal x-ray analyzer blade of a silicon crystal at each measurement point in an image. A phase shift of 360 degrees correlates to the translation of this blade by 0.31356 nm (for a 111 lattice spacing) [2]. The analyzer blade has been machined into the crystal as part of a flexure mechanism so that it can be translated by the application of a small force. The intent is to push the flexure with a thin, compliant fiber. This requires contact detection followed by precisely translating the analyzer blade flexure. To achieve multiple fringe translations with precise phase control, the translation stage is to be capable of a linear motion of the analyzer blade of 3-5 nm so that with control of around 1 part in 1000 the analyzer blade can be translated with picometer resolution.

KINEMATIC MOUNT

For accurate and repeatable placement of the x-ray interferometer monolith, a kinematic mount is utilized. Three grooves are cut in the base of the mount and three precision half-spheres are mounted to the base of the interferometer monolith [3]. Because the half-spheres are to be mounted directly to the interferometer monolith, other monolith structures can use the same mounting mechanism.

Custom plastic safety pieces will be placed around each monolith structure to provide additional security. These rapid prototyped

pieces, as marked in Figure 3, are installed to prevent the delicate silicon (or other crystal) structure from being knocked from the kinematic mount system.

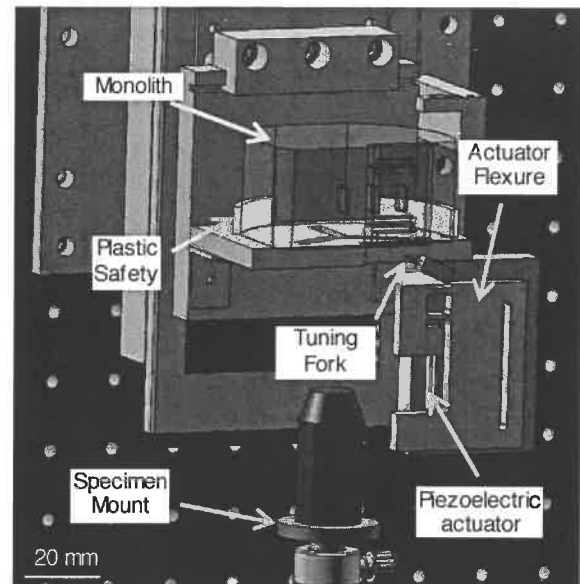


FIGURE 3. Detailed CAD of system design.

An adjustable leaf spring will be utilized to apply a controlled force to the monolith while on the kinematic mount. At the end of the spring metal, a half-sphere is used to distribute the force and thereby reduce surface stresses at the contact.

The mount assembly is easily removable, enabling precise replacement of arbitrary monoliths. This requirement of precision replacement is also used to provide coupling of the actuator system to the mount assembly.

ACTUATOR SYSTEM

To produce the 3-5 nm translation motion while also sensing initial contact with the flexure, a tuning fork oscillator will be utilized (Figure 3). The tuning fork is, in turn, mounted to a piezoelectric actuated flexure with a translation range of 15 μm . Hence it is necessary to provide a lever attenuation of around 3000:1. To achieve this, a tungsten or glass fiber will be added to a tine of the tuning fork oscillator with its stiffness designed to be approximately 1/3000 of that of the silicon flexure, see Figure 4.

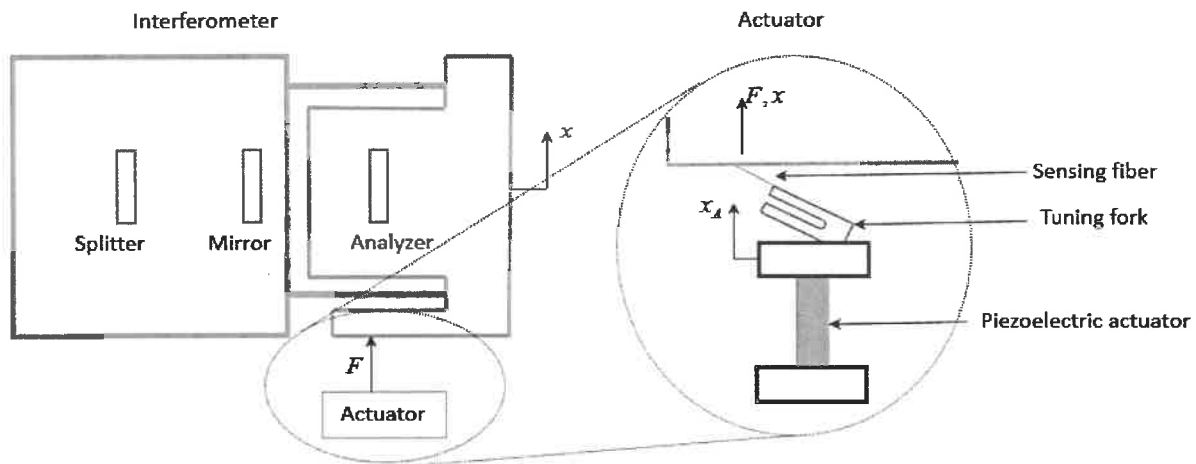


FIGURE 4. Schematic diagram of proposed driven mechanism.

ACTUATOR FLEXURE

To position the tuning fork, the longer range piezoelectric actuator is mounted into a 'four bar' monolithic flexure. Upon activation of the piezoelectric actuator, the flexure will move the tuning fork actuator towards the monolith while the tuning fork acts as a sensing element to detect when contact is made with the analyzer blade flexure, as seen in Figure 4.

A four-bar flexure was selected so that the specimen mount, as seen in Figure 3, can be safely placed between the splitter and mirror blades on the monolith structure without interference with other mechanisms necessary for specimen positioning and scanning.

The flexure is powered by a 3 mm x 4 mm x 20 mm piezoelectric actuator, which creates a force range of approximately 50-200 N. Using the

modeling software's built-in FEA tool, a 100 N load was applied to the flexure that produced a movement range of approximately 24.5 μm and showed the maximum stress was within yield strength for 6061 aluminum. The flexure design enables the tuning fork to be positioned before the vibration sensing is activated.

TUNING FORK SENSOR

A quartz tuning fork, commonly found in wrist watches, will fulfill two roles. Being a piezoelectric material, the tuning fork has the property that when electrically excited, the tines experience a corresponding mechanical deformation. Polarity of the electrical field changes the direction of deformation resulting in expansion or contraction of the tines.

When an AC signal is supplied to the tuning fork, the tuning fork will oscillate. By adjusting the frequency of the AC signal, the tuning fork can

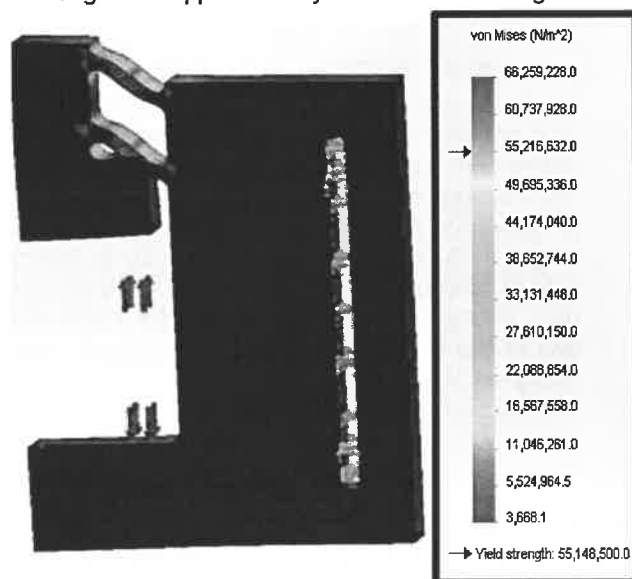


FIGURE 5. Stress measure for actuator flexure.

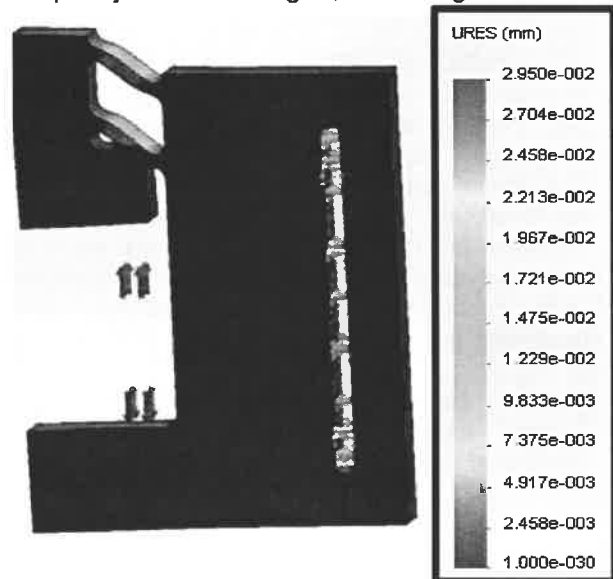


FIGURE 6. Displacement measure for actuator flexure.

be tuned to oscillate at its first modal frequency. While the tuning fork is used in oscillation mode, monitoring the electrical response will provide active contact detection. As an object comes into contact with the tuning fork while oscillating, the change in work done by the oscillator is seen in the electrical feedback.

Once the fiber is in contact with the analyzer blade flexure, the piezoelectric actuator is used to provide translation. In order to attenuate motion of the piezoelectric actuator to a displacement of 3-5 nm at the analyzer blade, a fiber will be added to the tine of the tuning fork closest to the analyzer blade flexure.

Previous tests have shown that when a heavy fiber is adhered to only one tine, the tuning fork loses sensitivity to the point of providing no contact data. It has been hypothesized that this is due to a loss in the symmetry of the tuning fork, resulting in a low resonant amplitude and correspondingly low sensitivity to contact. To compensate for this effect, a counter-mass will be added to the opposite tine.

RAPID PROTOTYPING

To determine the flexibility of the x-ray interferometer system, rapid prototyping is currently being used to build models of complete drive mechanisms. By having a fully rapid prototyped system, new interferometer designs will be tested prior to manufacturing and installing of the fragile and expensive single-crystal monoliths. New mounts for additional specimen types will also be evaluated on the rapid prototype test bed before implementing into the x-ray interferometer system.

CONCLUSION

To achieve project goals, three major mechanism designs are in development with these being:

- Adjustable mount for placement of the interferometer sub-system into existing x-ray facilities.
- Precision repeatable relocation of crystalline monoliths into the interferometer sub-system.
- Coupled and detachable translation system for replaceable sub-atomic analyzer blade motion control.

The bulk of the research focuses on developing the facility for a commercially viable x-ray

interferometer system. Current studies focus on the design and rapid prototyping of the entire interferometer subsystem, the fiber attenuator attachment to the tuning fork actuator, and performance evaluation studies.

ACKNOWLEDGEMENTS

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COMPLIANT-MOUNT AIR BUSHING SYSTEM FOR LOW-FRICTION MOTION OF AN INCLINED-PLANE IMPACT TESTING APPARATUS

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INTRODUCTION

A testing apparatus has been developed to investigate the deformation of tissue during impact and penetration by a medical needle at varying velocities. The apparatus acts as a variable-angle inclined plane, with the needle mounted on a carriage which accelerates towards impact due to gravitational force. As the needle penetrates tissue, the tissue deforms due to forces applied by the needle, which are highly dependent on both the position and velocity of the needle [1]. As a result, it is critical to characterize external forces acting on the carriage in order to isolate and evaluate the tissue reaction forces which will be compared to various theoretical models for needle-tissue interaction force.

BUSHING SYSTEM DESIGN

In order to minimize external forces acting on the carriage during travel, a system of air bushings is employed (NewWay Air Bearings S301201). The bushings travel on a pair of parallel-mounted 0.5 inch diameter stainless steel shafts. The lower shaft supports the primary weight of the carriage through two co-axially mounted bushings, constraining motion in translation along and rotation about both axes radial to the axis of travel. The third bushing travels on the upper rail, acting as an outrigger to constrain rotation about the axis of travel.

To accommodate shaft bending, minimize shaft diameter and weight, and allow for alignment errors during assembly, a compliant mounting strategy is employed for the bushings. The two primary bushings are mounted to the carriage on elastomeric columns, which are stiff in compression but allow some rotation and shear deformation. The outrigger bushing is mounted to the carriage only by its air supply lines, oriented to provide significant stiffness only in rotation about the travel axis. Figure 1 shows the carriage mounted on the rails at a substantially horizontal incline angle.

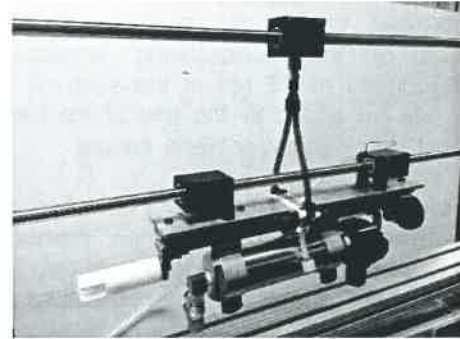


FIGURE 1. Test apparatus needle carriage featuring compliant bushing mounting strategy.

As such, frictional forces due to bushing misalignment are averted. However, the bushing system still experiences some frictional losses due to fluid shear within the gap between the bushings and the shafts. Much work has been done to model the complex fluid interactions within porous air bearings [2, 3]. An alternative first-order model has been developed to characterize the proposed bushing system.

Assuming steady flow, the frictional force acting on a bushing may be derived from standard Couette flow as

$$f_b = \frac{\mu l \pi d_b v}{h} \quad (1)$$

Where μ is the fluid viscosity, l is the length of the bushing, d_b is the inner diameter of the bushing, v is the velocity of the bushing with respect to the shaft, and h is the gap thickness equal to the radial bushing clearance. Since the outrigger bushing does not bear the weight of the carriage, its load is substantially less than the other bearings and it is considered to be coaxially aligned with the upper shaft; as such (1) is a suitable expression for the frictional force experienced by this bushing.

As the two primary bushings support the weight of the carriage, the bushings will deflect radially

out of co-axial alignment with the lower shaft. As such, the air gap between the shaft and the bushings is of non-uniform thickness. The relationship between the minimum radial gap thickness of one bushing and radial load is given by the manufacturer as

$$\delta = 564.286 - \sqrt{18275.5 + 14285.7 \cdot P} \quad (2)$$

Where δ is the radial deflection, measured in micro-inches, and P is the radial load. Both primary bushings deflect under the radial component of half of the carriage load, which changes as a function of incline angle as

$$P = \frac{mg \cos \theta}{2} \quad (3)$$

Where m is the mass of the carriage, g is the gravitational constant, and θ is the incline angle. To determine the force of friction, the fluid shear force is integrated over the varying gap height between the shaft and the bushing. Assuming that the compliant mounting method allows each bushing to remain axially parallel to its shaft despite radial misalignment, the force of friction felt by each primary bushing is given as

$$f_p = \int_0^{2\pi} \frac{\mu l \pi d_b v}{\frac{d_b}{2} - (\delta \cos \varphi + \sqrt{r_s^2 - \delta^2 \sin^2 \varphi})} d\varphi \quad (4)$$

Where r_s is the radius of the shaft. As such the total frictional force acting on the carriage during travel is given as

$$F_f = 2f_p + f_b \quad (5)$$

This expression of friction may be included in the differential equation of motion of the carriage, which is acted upon both by gravitational acceleration as well as the velocity-dependent frictional bushing force and is given as

$$m\ddot{x} + \Phi\dot{x} - mg \sin \theta = 0 \quad (3)$$

Where x is the displacement of the carriage in the direction of travel, Φ is the friction coefficient, defined by (2-5) and found to be constant for a given angle of incline. This equation may be solved to determine the velocity of the carriage at the moment of impact, given as

$$\dot{x}(t) = \frac{e^{-\frac{\Phi t}{m}} \left(-1 + e^{\frac{\Phi t}{m}} \right) \cdot mg \sin \theta}{\Phi} \quad (5)$$

TESTING

To validate the proposed model, the carriage was released from rest at various incline angles. The velocity was measured at impact by a beam-interruption photogate sensor. Figure 2 shows the theoretical and measured impact velocities at various incline angles.

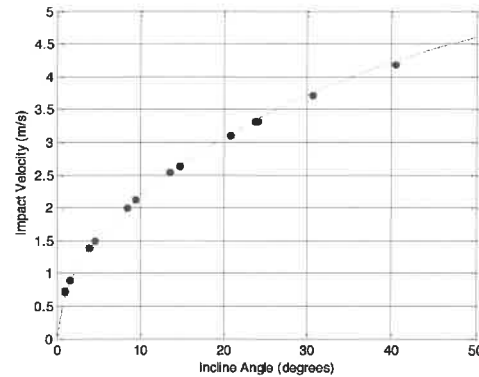


FIGURE 2. Theoretical (line) and measured (discrete points) impact velocity at impact.

The average error between the measured and theoretical impact velocities is 1.25%. This is mostly due to the errors at very low incline angles below 1 degree, at which the change in shaft angle due to beam bending significantly affects the acceleration of the carriage. At angles above 1 degrees, the average error is only 0.07%.

CONCLUSIONS

A compliant-mount bushing system allows for low-friction motion of the proposed test apparatus, and may be characterized to determine the effects of fluid shear friction. Although the specific mechanisms of bushing friction are highly complex, this work has discovered a first-order model which allows calculation of impact velocity to within 0.07%.

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LARGE STROKE SCANNING SYSTEM FOR FEMTOSECOND MICROMACHING OF CUSTOM OPTICAL STRUCTURES IN OPHTHALMIC MATERIALS

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BACKGROUND

Recently, a technique has been developed to write customizable, three-dimensional refractive index (RI) changes into ophthalmic materials using femtosecond laser micromachining. This was first demonstrated in ophthalmic hydrogels [1] where much larger RI changes were achieved than in other materials, such as glasses and plastics. The RI changes are caused by nonlinear absorption, which only occurs in the focal region of a beam focused using a high numerical aperture (NA) objective ($NA > 0.6$). Since the process only occurs in the focal region of a tightly focused beam, changes can be made with a lateral resolution of approximately $1\text{ }\mu\text{m}$ and an axial resolution on the order of $3\text{ }\mu\text{m}$. The magnitude of the change is dependent on the amount of energy deposited into the focal region, which is a function of the intensity of the laser light in the focal region and the translation velocity of the focal volume through the material.

By varying the translation velocity of the focal volume to create a specific RI profile, it has been demonstrated that it is possible to create lateral gradient index (GRIN) lenses in hydrogels [2]. In addition to ophthalmic hydrogels, femtosecond micromachining to change RI has also been demonstrated in live cornea where it is called Intra-Tissue Refractive Surgery (IRIS) [3]. More recently, it has also been demonstrated that it is possible to write GRIN lenses into the cornea of living cats and that such structures persist for at least 12 months [4].

The requirements for writing visual quality structures that cover the entire pupil of the eye put high demands on the scanning system that translates the focal volume. The first demand is the high NA that is mentioned above. This leads

to the $1\text{ }\mu\text{m}$ diameter spot. To write a full GRIN lens, it is necessary to write lines of RI change as closely spaced as possible without excessive overlap. This places a tight constraint on the positioning accuracy necessary. Also, because the RI change is a function of the intensity in the focal volume and the translation velocity of the focal volume, both those parameters must be controlled. All this must be done over an 8 mm diameter circular area to cover the entire pupil of the eye.

Over the course of the work mentioned above, several different scanning systems have been used to scan the focal region through the samples. One method was a galvanometer, using two mirrors located at the entrance pupil to the optical system to change the angle of the light entering the system, thereby changing the position of the focal region within the focal plane. This method allows for very high position accuracy and velocity control but is limited to the object size of the objective (generally less than 300 micrometers diameter). While such a small object size can still be useful for some tests, such as determining what actually changes in the material, it is more difficult to write a full 8 mm structure because of the complicated stitching necessary to write the 8 mm diameter. Additionally, stitching boundaries may present some scattering, affecting visual quality. Another method was using two orthogonal piezo-stepper stages to translate the sample instead of translating the focal volume. This gave the requisite scanning size but it was discovered that the stages used had significant higher frequency variations in scan speed resulting in high frequency RI fluctuations that significantly diminish the optical quality resulting structures [5]. Also, the setup required the

sample to be moved at high velocities which is impractical for clinical applications.

The most recent method used was a flexure-based system that used an objective attached to a carriage driven by a voice coil (VC) motor and guided by flexures. The first iteration of this scanning modality used a commercial vibration exciter as the flexure-VC assembly. This provided much better velocity consistency at the expense of control due to the low frequency cutoff of the system. This first prototype also did not have the requisite stroke, but greatly improved results were achieved in living cat cornea using that system [4]. To correct the issues with the first prototype but continue with the advantages of a flexure-based system, a custom flexure stage was designed and characterized [5].

CUSTOM SCANNING SYSTEM

The custom scanning system that was designed and built has two major components. The first is the mechanical scanner comprised of the flexure stage and two commercial stages to achieve the remaining two degrees of freedom. The second major component is an intensity control module comprised of an acousto-optic modulator and focusing optics.

Mechanical Scanner

The mechanical scanning head is composed of an objective attached to a moving carriage. The moving carriage is guided by folded parallelogram flexures. The folded parallelogram design was chosen to limit the size of the footprint and provide for a flatter profile that enables easier beam routing. The carriage is driven by four VC motors mounted inside the carriage with two pushing and two pulling at any given time. The flexure and VC assemblies are mounted to a vertical axis stage (Newport GTS30V) which is in turn mounted to a linear stage with its axis of translation perpendicular to that of the moving carriage. The scanner is shown in Fig. 1.

For each mechanical axis, it is necessary to have four degrees of freedom for aligning the laser beam. This results in a pair of kinematic mirror mounts for each turn of the optical axis to ensure that the optical axis remains parallel to all of the respective mechanical axes. This is important to limit beam walk when entering the objective which would result in inconsistent

performance throughout the range of motion of each of the mechanical axes.

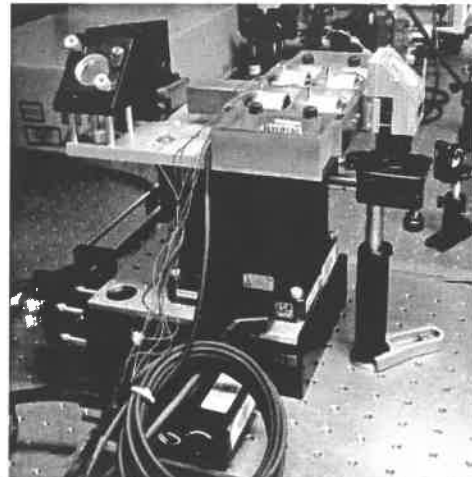


FIGURE 1. Scanner assembly with attached microscope objective.

The custom flexure stage was characterized both with and without an NA = 0.7 objective attached. The measurements were done using a custom heterodyne interferometer that used a beam reflected from a mirror that was epoxied to the stage [6]. The characterization setup is shown in Fig. 2.

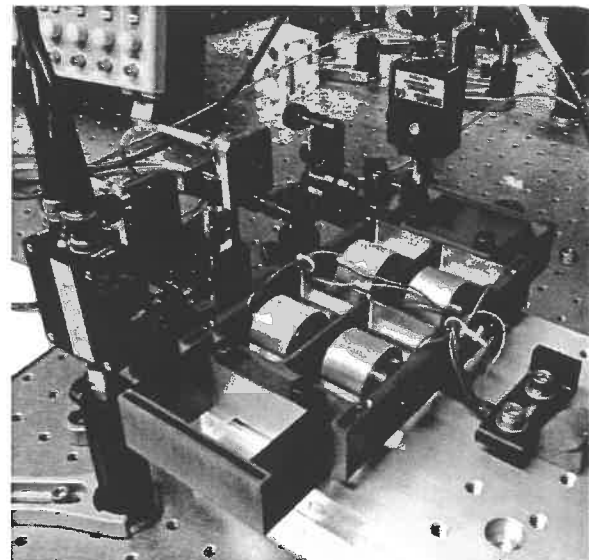


FIGURE 2. Interferometer setup used to characterize the flexure.

Using the setup shown in Fig. 2, gain curves of frequencies from 1 Hz to 50 Hz were measured. The natural frequency of the flexure from these gain curves was measured to be 26.6 Hz without the objective attached and 18.0 Hz with the

objective attached [7]. It was also discovered that when driven at lower frequencies and higher input voltage amplitudes the higher odd harmonics would start to appear in the waveform. As shown in Fig. 3, only the odd harmonics grow over time. If the frequency of the third harmonic is past the -3 dB cutoff then the resulting displacement of the flexure is purely sinusoidal.

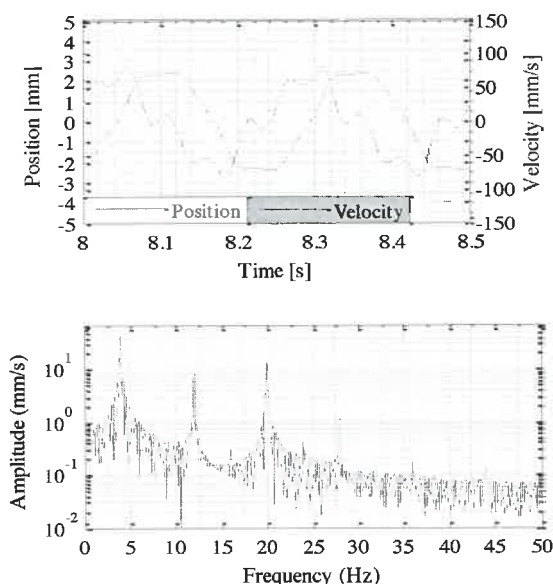


FIGURE 3. Sample displacement and velocity response with corresponding Fourier Transform to sinusoidal input of flexure with objective attached. Input is 4 Hz with an amplitude of 1.5 V. Measurement was taken after six minutes of continuous operation.

We believe that the distortion of the waveform evident in Fig. 3 is caused by thermal distortion of the flexures which comes to steady state after approximately 5-6 minutes of continuous operation. The waveform starts purely sinusoidal when the flexure first starts to oscillate. The thermal distortion is caused by the large amount of heat dissipated into the coils which are attached to the moving carriage. This design was chosen as opposed to attaching the field assemblies to the moving stage because we were trying to maximize the natural frequency and the field assemblies are four times more massive than the coils. For future iterations, either the orientation of the field and coils assemblies must be flipped or a thermal management system must be employed.

This flexure system is promising for our intended application where a stroke >8 mm is needed.

We have measured a maximum stroke of 9 mm with no high frequency velocity variations. We have written some preliminary structures in hydrogels, one of which is shown in Fig. 4. The structure shown in Fig. 4 was written using a constant laser intensity and demonstrates the size of structures that can be written with this flexure. There is some evidence of damage at the edges where the flexure must slow down to turn around. This will be eliminated using the laser intensity control system described below.



Figure 4. Hydrogel sample with an 8 mm x 8 mm pattern written in it without intensity control. Bright field image at 0.63 magnification.

Intensity Control

Originally, patterns were written using just a shutter to eliminate the turnaround points where the material could be burned due to the accumulated thermal energy. The refractive power from those structures was achieved with just the RI change from the changing translation speed of the first prototype of the flexure. This allowed for a structure with power, but does not allow for control of the structure being written. To completely control the RI changes that are written into the materials, it is necessary to also control the intensity of the laser beam throughout the scanning process. This has been achieved using the first diffracted order from an acousto-optic modulator (AOM).

In either case, the simple shutter or the AOM for complex intensity control, it was necessary to calibrate the timing of the system to account for the mechanical phase delay in the flexure. This was done using a position sensing detector (PSD) and a photodiode setup as shown in Fig 5.

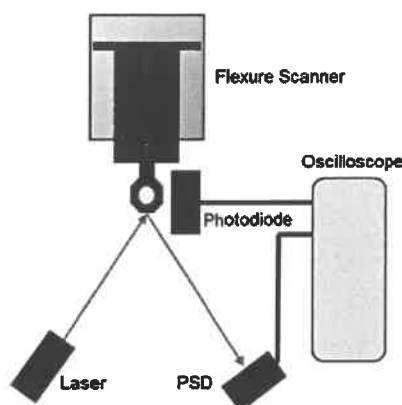


FIGURE 5. Setup for calibration of shutter/AOM timing relative to flexure motion.

The PSD was used to determine the timing of the flexure while the photodiode measured whether the laser was passing through the objective. The signals from both were measured using an oscilloscope to compare the timing. A sample oscilloscope reading is shown in Fig. 6. The phase delay of the signal passing to the shutter or AOM was then adjusted to such that the times during which no light was passing through the objective were centered on turnarounds of the flexure.

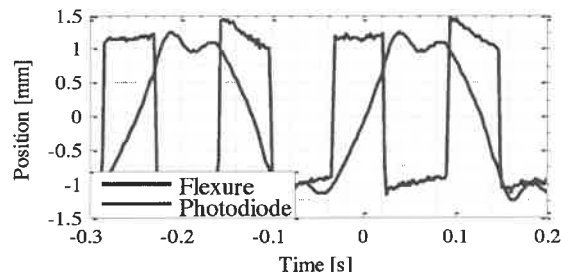


Figure 6. PSD and photodiode signals used to calibrate shutter/AOM. The low signal section of the photodiode is shifted using a delay on shutter/AOM to position it over the turnaround points.

The above procedure is capable of obtaining a rough phase delay, but refining the timing necessitated writing structures and measuring the offset to achieve the accuracy necessary to write highly accurate phase structures. This was done simply by writing structures with differing AOM delays and measuring the offset of the upstroke and down-stroke. Figure 7 shows a sample of the pictures that were used to perform this calibration.

As shown in Fig. 7, we were able to achieve an offset of less than $5\text{ }\mu\text{m}$. This is sufficient for our current research. In the future, we will likely need to improve this but that should be straightforward since we are not at the sampling limit of our program yet.

CONCLUSION AND FUTURE WORK

With the AOM calibrated to the movement of the flexure, it is possible to write any arbitrary intensity profile across the entire region of writing. This will allow not only the correction of optical power and astigmatism in the eye but also possibly of higher order aberrations as well.

FIGURE 7. Picture of micromachined hydrogel used to calibrate the AOM. The timing delay on the AOM signal was adjusted to minimize the position offset between the upstroke and downstroke of the flexure.

Currently the AOM and flexure are operated in open loop, which should be sufficient to write basic patterns. The capability to measure the movement of the flexure in real time was included in the design but has not been implemented due to limitations in signal processing at high frequencies. Once implemented, it will allow closed loop control so that the AOM can correct for any velocity departures from a sinusoid, such as those demonstrated earlier in Fig. 3.

ACKNOWLEDGEMENTS

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ANALYSIS OF TEMPERATURE DISTRIBUTION IN BALLSCREW SYSTEMS BY THE FINITE ELEMENT METHOD

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ABSTRACT

Finite element (FE) method is applied to study temperature distribution analysis in ballscrew systems. For accurate modeling and analysis, heat generation mechanism of ballscrews as well as temperature model is accurately derived through parametric modeling and thermal contact conductance (TCC) analysis at the nut-shaft interface. To verify the developed FE model, experiments are conducted for key temperature measurements.

INTRODUCTION

As ballscrews have high efficiency, stiffness and precision, they have been used for precision motion transfer elements in the machine tool and manufacturing industries. However, friction torque at the nut-shaft interface generates heat during working. Thermal deformation error due to the heat deteriorates positioning accuracy of ballscrew-driven servos. In order to compensate for the thermal error without using extra linear scales, accurate estimation of the thermal expansion and its thermal stiffness is required for precision positioning [1-4].

To calibrate the thermal expansion of the ballscrew accurately without using extra sensors, whole temperature measurement of the ballscrew according to assembly and operating conditions are required first. However, it is extremely inefficient and almost impossible to acquire the whole temperature distribution by measuring temperature at every point. Hence, a thermal error model estimating the whole temperature and error from the mathematical model according to assembly and operating conditions is required [2-6]. FE method and finite difference model (FDM) have been studied to solve above problems [2-3, 5].

In this paper, contrary to previous researches [3, 5], a parametric FE model is devised to analyze

temperature in the ballscrew systems. Heat generation mechanism of ballscrews as well as thermal analysis is studied accurately through parametric modeling and TCC consideration at the nut-shaft interface. To verify the developed FE model, experiments are conducted through temperature measurements on the test rig. ANSYS parametric design language (APDL) tool is applied for the FE analysis. Ball slip, viscous friction and rolling contact friction effects, as well as natural and forced convection coefficients of the ballscrew are applied through the developed thermal characteristics identification method [4].

THERMAL ANALYSIS

Fig. 1 shows an axisymmetric finite element model of the ballscrew system. Thermal and structure coupled 4,500 elements with about 1mm mesh size is applied for FE analysis. To simulate the feed rate, reciprocating motion and stroke, APDL is applied as well. Temperature distribution according to operating conditions is computed on the following assumptions:

- i) Screw shaft is a solid cylinder.
- ii) Heat flux generated at the nut-shaft interface and support bearings are uniform between sample intervals.
- iii) Convection coefficient is uniform during constant feedrate and changes according to moving speed.

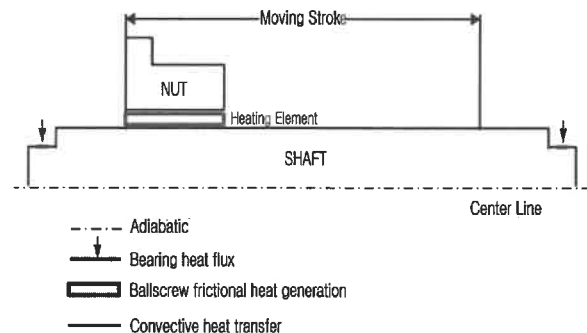


FIGURE 1. Axisymmetric finite element model.

- iv) Balls and grease between the nut and the shaft are simplified as a heating element. TCC is uniform and depends upon preload and thrust force, and specific heat is 0.

In the ballscrew systems, support bearings generate frictional heat given by Eq. (1). Ballscrew nut-shaft interface generates frictional heat according to assembly and operating conditions by Eq. (2). Convection heat flowrate on the screw shaft is given by Eq. (3).

$$\dot{Q}_{br} = 1.047 \times 10^{-4} \omega T_f^{br} \quad (1)$$

$$\dot{Q}_{bs} = 1.047 \times 10^{-4} \omega T_f^{bs} \quad (2)$$

$$\dot{Q}_{conv} = h A^s [\theta(t) - \theta_a(t)] \quad (3)$$

where ω , T_f , h and A^s are rotational speed, friction torques, convective heat transfer coefficients and surface areas of the ballscrew shaft, respectively. Ambient temperatures θ_a are measurable disturbances.

In the FE model, heating element means rolling contact between grooves and balls of the ballscrew. Across the rolling joints at the nut-shaft interface, heat transfer reveals nonlinear behavior due to the preload and thermal conductance changes during operation. These phenomena means TCC is time-varying and multivariable across the interface. However, in this paper, TCC is time-invariant and constant on the nut-shaft interface. In addition, estimation of these variables is conducted through some modification of the previously developed inverse heat and key thermal characteristics estimation methods [1, 4, 6-7].

ESTIMATION OF KEY PARAMETERS

To obtain key thermal parameters according to the assembly and operating conditions, the experimental setup shown in Fig. 2 is used for the modified inverse analysis procedure [4]. Some thermocouples, torque sensors, heat flux sensors, and a data acquisition system is applied for the inverse heat analysis. To estimate TCC, drilling is used to make holes on the nut and shaft in axial and radial directions. Iterative estimation processes of heat fluxes as well as tuning procedures of the FE model have been conducted to obtain key parameters. Validity of the developed FE model is to be confirmed through experiments.

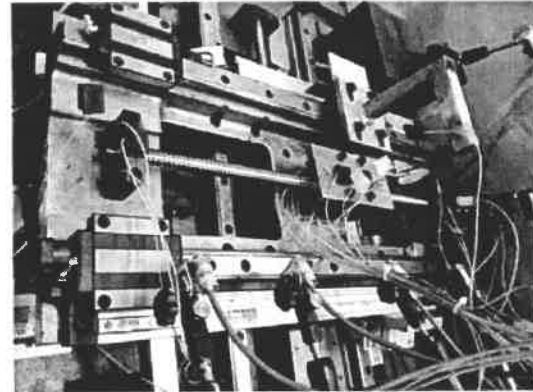


FIGURE 2. Experimental setup.

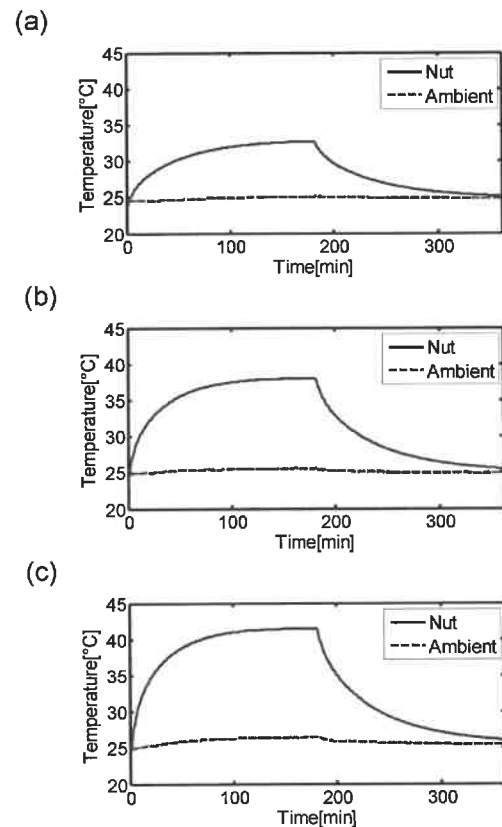


FIGURE 3. Measured nut temperatures at feedrates of (a) 1.25 m/min, (b) 2.5 m/min and (c) 3.75 m/min.

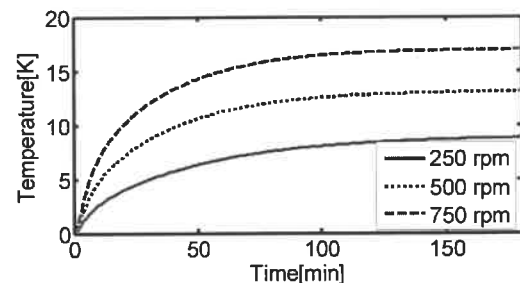


FIGURE 4. Nut temperatures rise.

EXPERIMENTS

Double-nut ballscrew with shaft diameter 16mm, lead 5mm, stroke 100mm and preload 760N is applied for experiments. Fig. 3 shows temperature measurements on the nut at feedrates of 1.25, 2.5 and 3.75 m/min for 3 hours rotation and 3 hours rest. Fig. 4 shows relative temperature rise of nut for 3 hours heating process. Up to 50 minutes, temperatures rise rapidly. After 50 minutes temperatures converge to steady state. This confirms that thermal deformation error is reduced if we use the machine after proper warm-up.

FE ANALYSIS

Fig. 5 shows temperature comparisons between experiments and FE analysis at feedrates of 1.25, 2.5 and 3.75 m/min. They show good agreement. Fig. 6 shows simulated temperature distributions for 3 hours reciprocation. Maximum temperature occurs at the nut-shaft interface during moving in the ballscrew system. These confirm that the devised FE model of the ballscrew system is accurate to simulate temperature distribution of ballscrew systems.

CONCLUSIONS

In this paper, FE temperature distribution analysis of ballscrew systems has been conducted through accurate modeling of frictional heat and TCC at the nut-shaft interface.

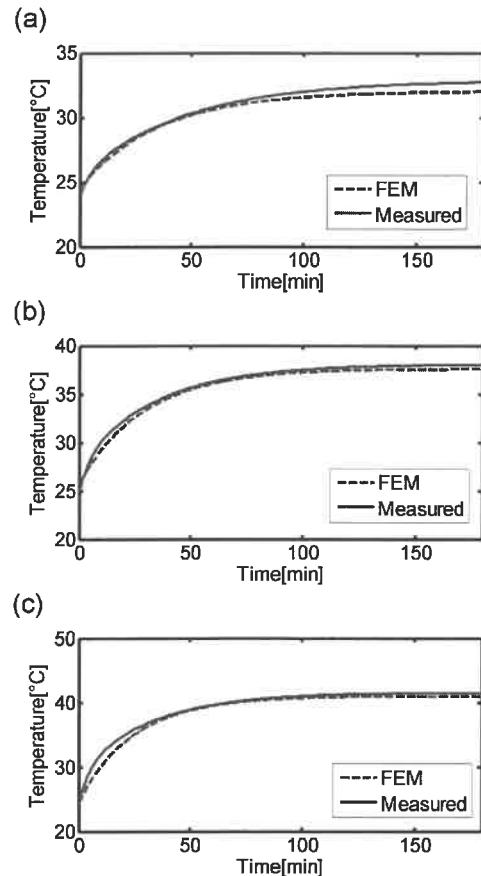


FIGURE 5. Measured and computed nut temperatures at feedrate of (a) 1.25 m/min, (b) 2.5 m/min and (c) 3.75 m/min.

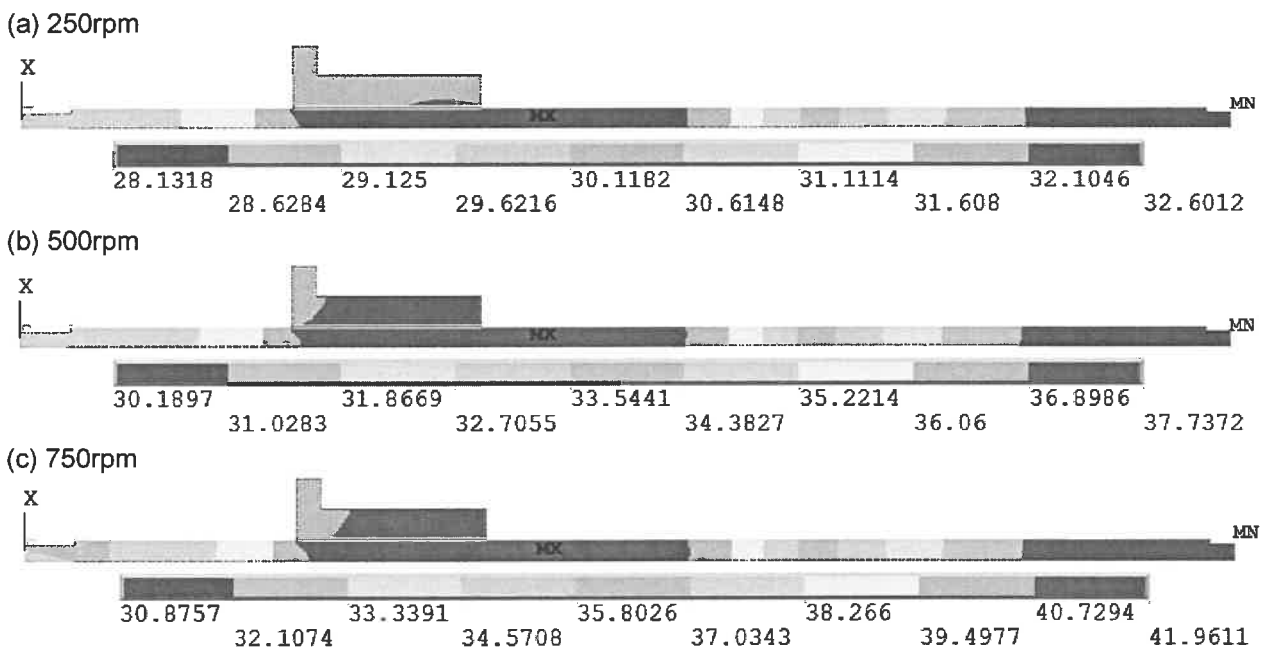


FIGURE 6. Temperature distributions in the ballscrew system.

Key thermal parameters such as convection heat coefficient, frictional torque and thermal contact conductance at the nut-shaft interface as well as frictional heat of support bearings are estimated through inverse heat analysis linked with experiments. To describe reciprocating motion in the moving stroke accurately, APDL tool is applied for the FE analysis. Experimental results confirm accuracy and validity of the devised FE modeling method.

ACKNOWLEDGEMENT

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THERMAL ERROR MODELING OF SHAPE ELEMENTS IN GIANT FLOOR TYPE BORING MACHINES

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ABSTRACT

Transient temperature distribution and thermal deformation of the shape element of a giant floor type boring machine are computed from finite element analysis (FEA). Using time series input and output data obtained from the FEA, thermal deformation transfer function (TDTF) is obtained through a black-box type system identification method. Applying the identified TDTF to the CNC, real-time compensatory error control is to be realized accurately. Validity of the proposed TDTF modeling is confirmed through FEA.

INTRODUCTION

Giant floor type boring machines are widely used in construction, oil field, wind energy, plant and shipbuilding industries. To make giant machining volume, they are made of long floor bed, huge column, boring and milling spindle assembled on the ram as shown in Fig. 1. Its machining accuracy depends upon deviation from the planned relative movement between the tool center point (TCP) and the workpiece. This relative error is called the volumetric error. It comes from joint errors of moving axes and shape deformations from the machine tool structures. As the stroke range is over several meters and the workpiece is huge, machining time takes several hours. Contrary to small and regular machine tools, volumetric error of the mega machine tool degrades machining accuracy severely. This error comes from non-repeatable joint and shape deformation errors according to internal and external heat sources. In previous researches [1-2], to estimate thermal deformation of the mega machine tool, kinematic error models were formulated with first order transfer functions. Thermal gains and time constants of the transfer functions were formulated through heuristic approach [2].

FEA has been applied to thermal design of small-size machine tools. Haitao, et al. applied FEA to study thermal behavior of a turning spindle [3]. Josef Mayr, et al. applied FDM-FEA

to analyze thermo-mechanical machine tool behavior [4]. Modeling of thermal behavior of a two-axis tilting rotary table for multiaxis machine tools was studied by the finite element method for predicting machine tool error in heating process [5]. They have used the FEM models for possible change of design and modification of materials.

Contrary to such thermal behavior analysis and design of machine tools using FEM, real-time thermal analysis as well as compensatory control of machine tools through TDTFs and mathematical thermal deformation models are required for CNC machine tools [6-7]. However, to formulate such mathematical error models regarding the giant machine tool through experiments is very difficult, expensive and time consuming. Derivation of TDTFs according to internal and external heat sources analytically in the discrete-time domain is required through data acquisition and fitting of input and output data without experiments.

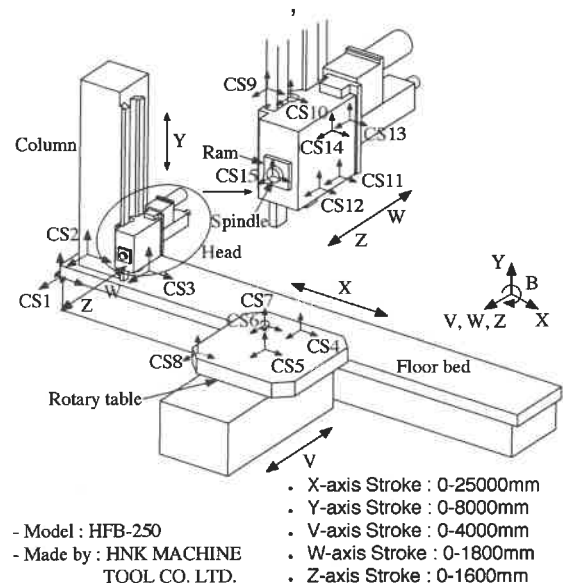


FIGURE 1. Coordinate systems of the floor type boring machine.

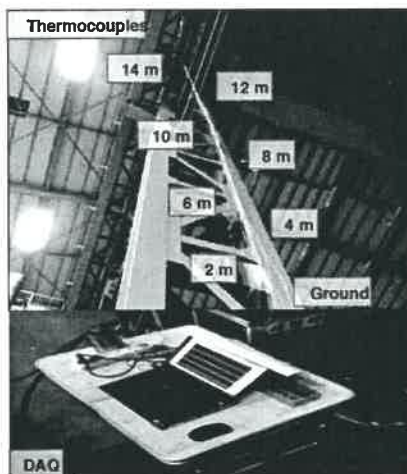


FIGURE 2. Ambient temperature measurement.

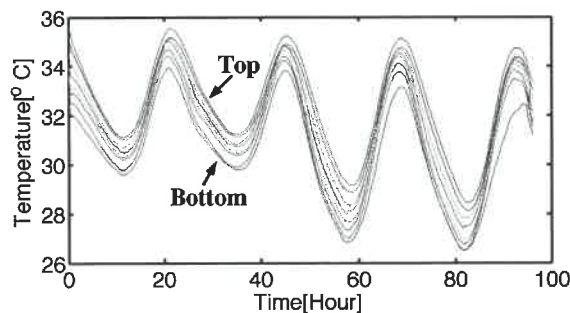


FIGURE 3. Ambient temperatures every 2m.

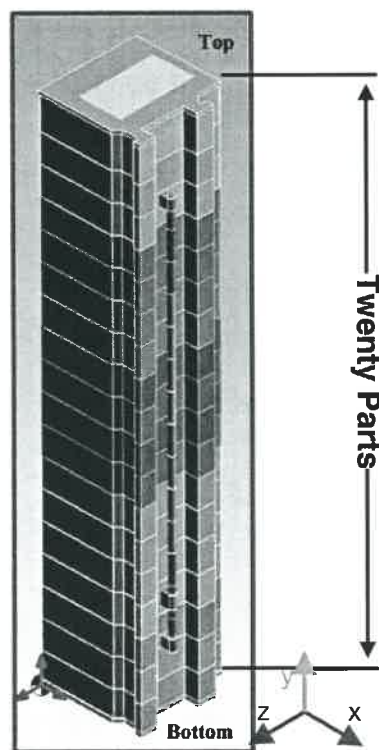


FIGURE 4. FE model of the column unit.

In this paper, to identify accurate TDTFs of the shape error elements according to ambient and internal heat inputs without experiments, derivation procedure of TDTFs for the column element of a giant floor type boring machine is conducted through FE analysis and black-box type system identification methods. ANSYS is applied to compute temperature distribution and thermal deformation of the column unit.

FEA OF SHAPE ERROR ELEMENT

Fig. 1 shows schematic diagram of the giant floor type boring machine. It consists of the floor bed, column, head, ram, etc. As the height of the machine is over 15 meters, thermal deformation behavior of the column is a dominant shape error element. For accurate thermal deformation modeling of the shape element in the column unit, ambient temperatures as external heat inputs are measured every 2m height above the ground level in the factory as shown in Fig. 2. Fig. 3 shows measured ambient temperatures for 96 hours. Temperature difference between bottom and top is 2~3°C. Diurnal range of 6~9°C generates time variant thermal deformation of the column. Heat flux generated from the head unit according to operating conditions acts as internal heat sources.

Fig. 4 shows a FE model of the column. For accurate thermal deformation analysis, it is divided into 20 parts. Ambient temperatures of each part is calculated by linear interpolation between the bottom and top temperatures. Fig. 5 shows transient thermal deformation results due to ambient temperature and operating conditions (12 hours heating and 12 hours cooling), respectively.

IDENTIFICATION OF TDTFs

TDTFs of shape error elements are composed of environmental and operating models. Using single-input and single-output approach, a specific height ambient temperature becomes input variable, and thermal deformation of the coordinate origin of the shape element becomes output variable for the environmental model. For the operating model, input is heat flux generated from the head unit, and output is the corresponding thermal deformation at the coordinate origin. By collecting input and output data according to sampling interval, and using discrete modeling method [8], dynamic thermal error model is obtained as shown in Fig. 6 and Fig. 7. Fig. 6 compares FEA and modeled TDTF

results regarding ambient temperature. Fig. 7 shows them regarding operating conditions when step type heat flux is generated from the head unit. Identified TDTF shows very similar deformation behavior comparing with the FEA results as shown in Fig. 6 and Fig. 7. Identified environmental TDTFs converted to continuous time domain in each axis are given by

$$\begin{aligned} TF_x^E(s) &= \frac{0.002014s + 8.547 \times 10^{-5}}{s^2 + 0.04101s + 5.08 \times 10^{-6}} \\ TF_y^E(s) &= \frac{0.0123}{s + 0.0006817} \\ TF_z^E(s) &= \frac{-1.803s^2 - 0.00173s - 4.181 \times 10^{-8}}{s^3 + 0.01535s^2 + 3.321 \times 10^{-5}s + 1.593 \times 10^{-9}} \end{aligned} \quad (1)$$

Operational TDTFs with respect to operating conditions of the machine tool in each axis are given by

$$\begin{aligned} TF_x^I(s) &= \frac{0.08966s^2 + 0.00551s + 4.034 \times 10^{-7}}{s^3 + 0.02932s^2 + 0.000901s + 4.31 \times 10^{-8}} \\ TF_y^I(s) &= \frac{0.0002523}{s + 7.73 \times 10^{-5}} \\ TF_z^I(s) &= \frac{0.1275s^2 + 5.216 \times 10^{-5}s + 3.921 \times 10^{-9}}{s^3 + 0.009986s^2 + 3.395 \times 10^{-6}s + 1.673 \times 10^{-10}} \end{aligned} \quad (2)$$

VERIFICATION OF TDTFs

To verify the identified TDTFs, a new 24 hours test procedure including both external heat source and internal heat source is proposed. External heat source is 24 hours ambient temperature in the factory. Internal heat source is composed of the 4h heating, 5h cooling, 7h heating and 8h cooling procedures. Fig. 8 shows thermal deformations estimated by the TDTFs and FEA processes. Maximum deviation of 15 μm occurs in X-axis. However, they match very well in each axis. This confirms the verification.

CONCLUSIONS

Formulation process of TDTF for the column unit of a giant floor type boring machine is conducted by using FEA. Ambient temperatures and heat fluxes from the head unit are applied as external and internal heat sources, respectively. Dynamic error models are derived from collected input and output variables through the black-box type discrete modeling technique. It is confirmed that identified TDTFs work very well for estimating time variant thermal deformation. The developed procedure is applicable to not only identification procedure of TDTFs during experiments, but

also real-time compensatory control of machine tool thermal errors.

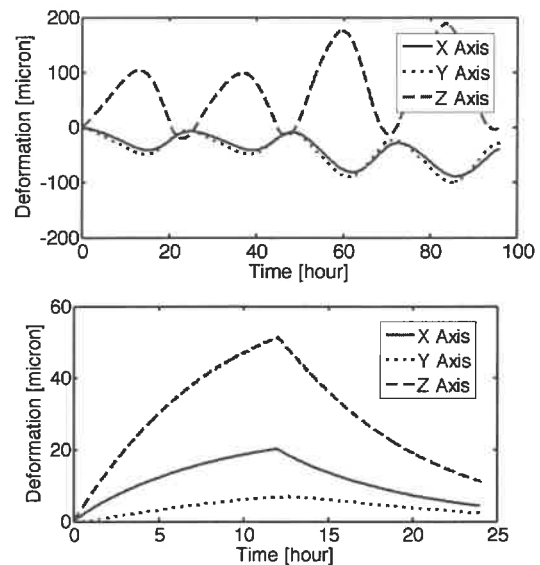


FIGURE 5. 3D thermal deformation according to ambient temperature (Top) and operating conditions (Bottom).

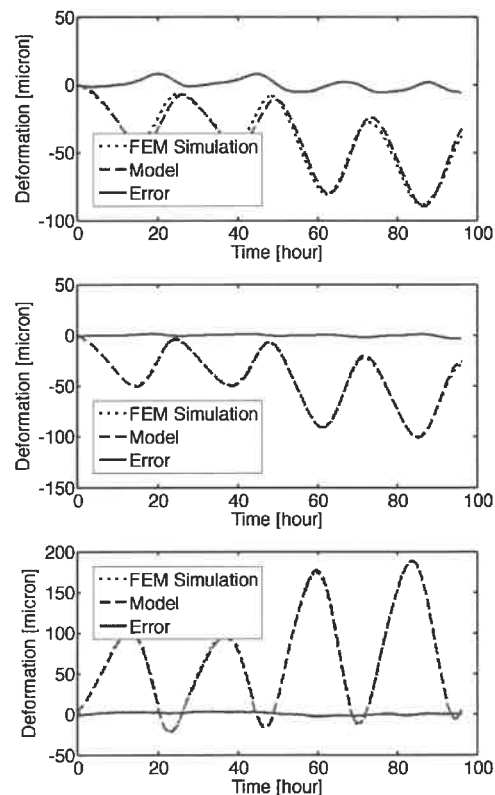


FIGURE 6. Comparisons of TDTF and FEA results regarding ambient temperature input in X, Y and Z Axes.

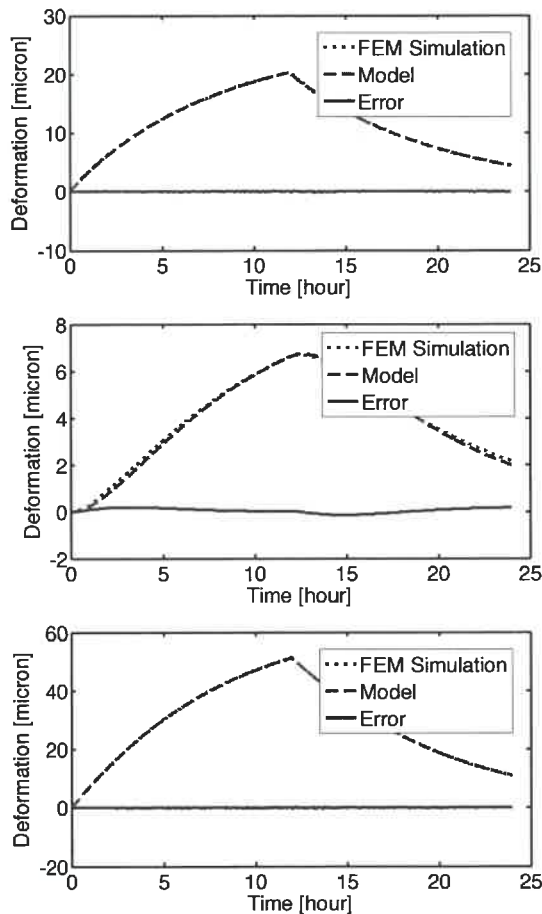


FIGURE 7. Comparisons of TDTFs and FEA results regarding operating conditions in X , Y and Z axes.

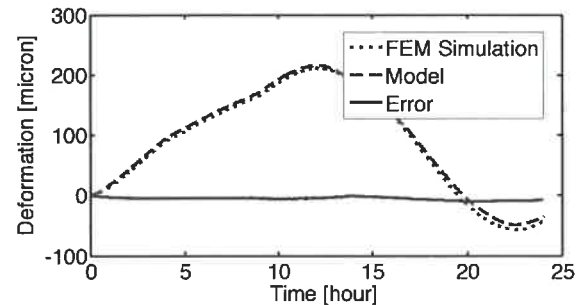
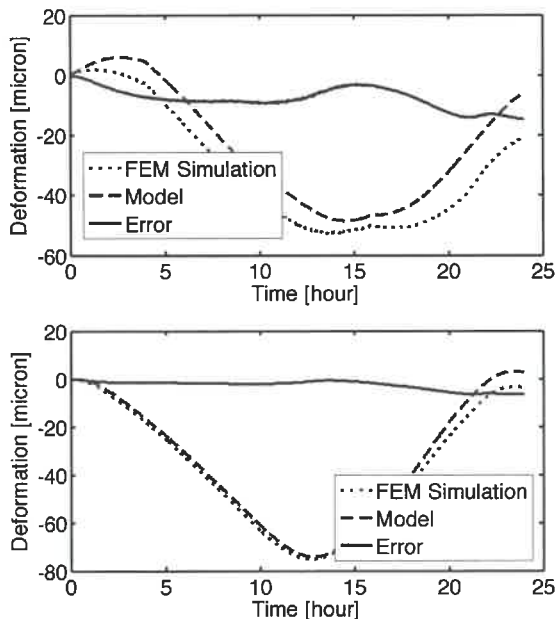


FIGURE 8. Verification procedure of identified TDTFs in X, Y and Z axes.

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Actuation Schemes for Flexible Hull, Undersea Vehicles

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INTRODUCTION

An investigation into the system level efficiency of flexible hull, undersea vehicles shows values as low as .12%, as estimated on page 114 of "Design of Biomimetic Compliant Devices for Locomotion in Liquid Environments". (Alvarado) This was found to be the result of resistive power dissipation in the actuators, which dissipated 99.45% of their input power. (Alvarado) The transmission mechanisms that control the motion of the propulsive hull were also estimated to dissipate 43% of their input power (Alvarado). This work investigates actuation schemes to minimize these losses.

To reduce the resistive power dissipation, an open loop, series-elastic drive was designed. This demonstrated 70% efficiency against a viscous load (Sun), however the hydrodynamic efficiency of such a scheme would be limited by its ability to track predefined swimming kinematics.

A direct drive Lorentz force actuation scheme was also studied, and proved to eliminate the need for intermediate power transmission elements. However, without the equivalent of a gearbox these actuators operated outside of their optimum operating point. This resulted in a modest efficiency (~10%) and a torque limit an order of magnitude too low. (Church)

The current configuration under study uses a high power rotary actuator to provide the majority of the power required for locomotion. This power is distributed along the tail through the use of passive elements. Anterior joint motion is supplemented by small, low power actuators at each joint. Experiments will be done to show that such a configuration allows for a reduction of transmission losses, and that the inherent efficiency of geared rotary actuators is maintained. Additionally, this scheme will allow for conventional control system design in order to track the kinematic trajectory of Thunniform swimming.

An efficient, flexible-hull, undersea vehicle would offer many benefits including non-invasive exploration of deep sea habitats, long distance underwater travel, discrete underwater travel, and high maneuverability.

BACKGROUND

To increase the efficiency of the flexible hull machine replicating the motion of Thunniform swimmers, a model of the interaction pressure between the water and the solid body (Eq 2) was integrated along the surface of the robot to derive the expected reaction moment on the rotary joints of the robot. This model was adopted from the analysis of Alvarado in "Design of Biomimetic Compliant Devices for Locomotion in Liquid Environments".

$$h = (c_1 + c_2 x + c_3 x) * \sin(w * t - k * x). \quad (1)$$

$$L_y = m * \frac{\delta^2 h}{\delta t^2} + 2 * U * m * \frac{\delta^2 h}{\delta t \delta x} + U^2 * m * \frac{\delta^2 h}{\delta x^2}. \quad (2)$$

Here "m" is the added mass of the surrounding fluid, "U" is the forward speed of the robot, "h" is the lateral displacement of the robot's spine, "x" is the position along the robot's spine, "w" is the swimming frequency, "k" is the wave number, and "L_y" is the linear pressure, assuming a uniform profile for the machine to simplify the analysis.

The reaction forces were analyzed using a free body diagram of the system which also accounted for the inertial effects of the four rigid links composing the machine, assuming perfect tracking of the desired motion. This resulted in a system of moment balance equations.

$$I_t * \alpha_t = M_{2,t} + T_{k3} + T_{b3} + (l_{cg,3} - l_3) R_{y3} + \frac{(l - l_3)}{2} \sin(\theta_3) T + \int_{l_3}^L (x - l_{3,cg}) L_y dx \quad (3)$$

$$I_2 \alpha_2 = -M_{2,t} + M_{2,1} + T_{k2} + T_{b2} - T_{k3} - T_{b3} + (l_3 - l_{cg,2}) R_{y3} + (l_{cg,2} - l_2) R_{y2} + \int_{l_2}^{l_3} (x - l_{2,cg}) L_y dx \quad (4)$$

$$I_1 \alpha_1 = -M_{2,1} + M - T_{k2} - T_{b2} + T_{k1} + T_{b1} + (l_2 - l_{2,cg}) R_{y2} - (l_{cg,2} - l_1) R_{y1} + \int_{l_1}^{l_2} (x - l_{1,cg}) L_y dx \quad (5)$$

$$I_h \alpha_h = -T_{k1} - M - T_{b1} - (l_1 - l_{cg,1}) R_{y1} + \int_0^{l_1} (x - l_{1,cg}) L_y dx. \quad (6)$$

This system of equations resulted in three expressions approximating the expected torque profile required at each joint to approximate the biological motions of swimming. Based on the phase between the required torques, the link angular positions and their angular velocities, damping and stiffness coefficients were selected for each joint to minimize the RMS power required by the actuators in our design. (See Figure 1)

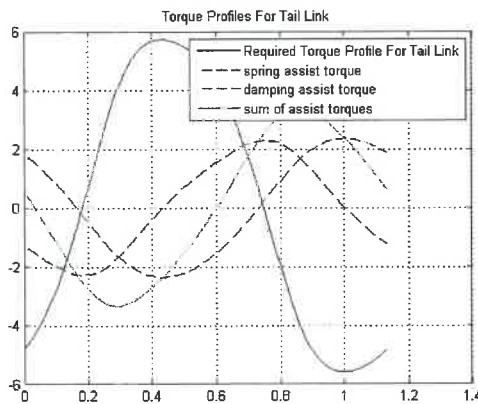


Figure 1: Plot of the expected torque profile along with a profile representing the phase of the possible spring torque and damping torque.

Based on the initial power estimates, we sized appropriate actuators and designed the structure based on their geometric constraints. The torque profile estimate and fluid-body interaction model will then be improved based on experimental data measured with the initial prototype, to be built shortly.

SERIES ELASTIC ACTUATOR

Results from the semi-passive configuration are forthcoming; however measurements of the efficiency of the series elastic actuator compared to those of the direct drive actuator show its potential superiority. (See Figure 2) (Sun, 2013)

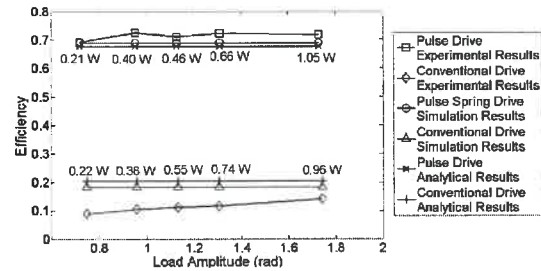
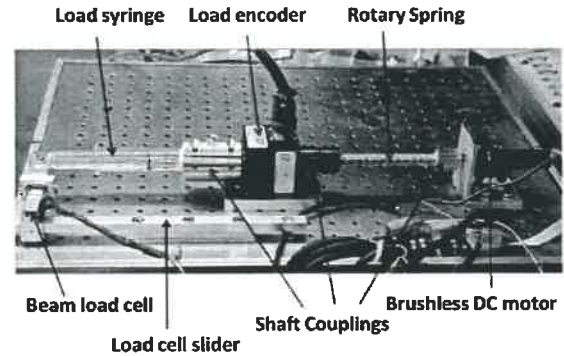


Figure 2: Top: This is an image of the experimental setup that Sun used to measure the efficiencies of a series elastic/pulse drive and a direct drive against a variety viscous loads. Bottom: Plots of the energy efficiency for both the pulse gear drive and the conventional gear drive. A single stage gearbox with a ratio of 4.4 was used for both the actuators. The pulse gear drive achieves approximately 70% energy efficiency while the energy efficiency of the conventional drive ranges from 10% to 15%. Taken from (Sun).

This shows that for the same position trajectory, a series elastic drive can potentially have energy efficiency significantly greater than that of a purely rigid drive. This is because the duration of the applied force is shorter for a series elastic drive, resulting in less resistive power loss in the actuator, as seen in Figure 3.

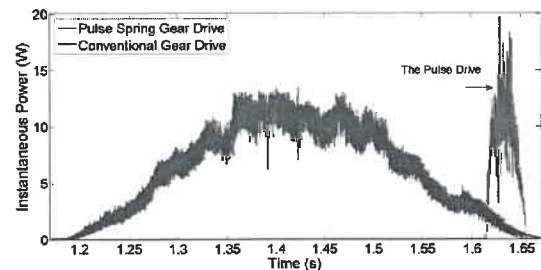


Figure 3: Plot of driver motor instantaneous power. Taken from (Sun).

This shows that the electromagnetic power applied to a series elastic drive has greater amplitude but shorter duration than the power applied to a rigid drive.

LORENTZ FORCE ACTUATOR

A direct drive actuation scheme was investigated as a baseline for future research. (Church, 2014) A highly scalable design for modular Lorentz force actuators was developed. The actuators were designed to directly drive tail foil sections, or vertebrae, in an oscillatory motion to provide thrust. The design process was automated to facilitate implementation in different sized vertebrae. This automated design process was used to define the dimensions of the actuators implemented in the test setup shown in Figure 4, obtained from Church's "Modular Lorentz Force Actuators for Efficient Biomimetic Propulsion of Autonomous Underwater Vehicles".

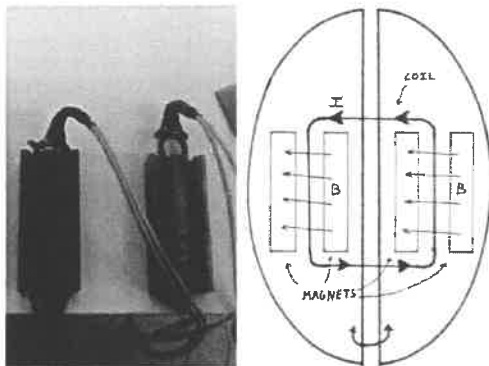


Figure 4: Top: The test setup used to measure the efficiency of the Lorentz force actuators. Bottom left: The optimal actuator design as determined by the automated process. This design resulted from a script that considered the geometric constraints of the housing vertebra, a finite element magnetic model of the field strength, and a table of available magnets

differentiated by their various dimensions. Bottom right: Schematic depiction of distributed Lorentz force actuators in a single vertebra. (Church)

The test stand pictured above was constructed to evaluate both the static and dynamic performance of the actuators. The prototype actuators achieved the required motion and demonstrated modest performance at a variety of load levels, see Table 1. (Church)

Damping $\left(\frac{N \cdot m \cdot s}{rad}\right)$	RMS efficiency
0.091	1.4%
0.050	2.3%
0.012	10%
0.0067	16%

Table 1: A table of the viscous loads used to test the dynamic efficiency of the actuators.

The Lorentz force actuators were designed to be constrained by only the frame of the robot. This clever choice of constraint eliminated the transmission elements between the actuator and the robot, which also eliminated this loss mechanism. However, in order to operate at an optimally efficient surface speed given the predefined joint trajectory, some sort of transmission element appears necessary. This need is suggested by the modest efficiency of the 4 watt prototype actuators shown in Table 1. (Church)

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A WIND ENERGY PLAN THAT FITS AMERICA'S RESOURCES

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SUMMARY

The return on investment from utility scale wind turbines can be improved by a paradigm shift from Horizontal Axis Wind Turbines (HAWT) to Vertical Axis Wind Turbines (VAWT) **because** they may be made to float easily, allowing inexpensive siting in Class 6 winds rather than the class 3 or 4 wind near shore. Recent technology advances in high performance racing yacht designs can be borrowed and tension and compression principles, (like crossing a sail boat mast with a bicycle wheel) to build high strength, light weight structures. These factors more than make up for the lower efficiency of VAWT design and provide a doubling of the capacity factor.

Using radial air or water bearings to support the rotor at the perimeter, VAWT's are less expensive and faster to build, they would have just one moving part, the rotor. Direct drive generation at the perimeter may be employed to eliminate the whole drive line. Deep water siting

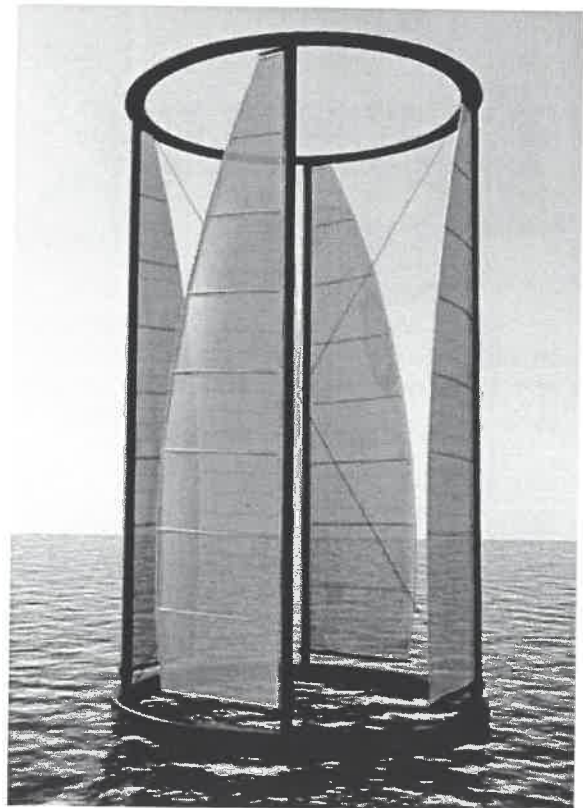
allows for several methods of energy storage (pumped hydro or buoyancy).

The turbine center of gravity is low with the generation components near the bottom of the structure, this makes for easy service at sea and the shallow draft allows complete erection at port and the ability to tow the turbine to sea or back to port in day for major service.

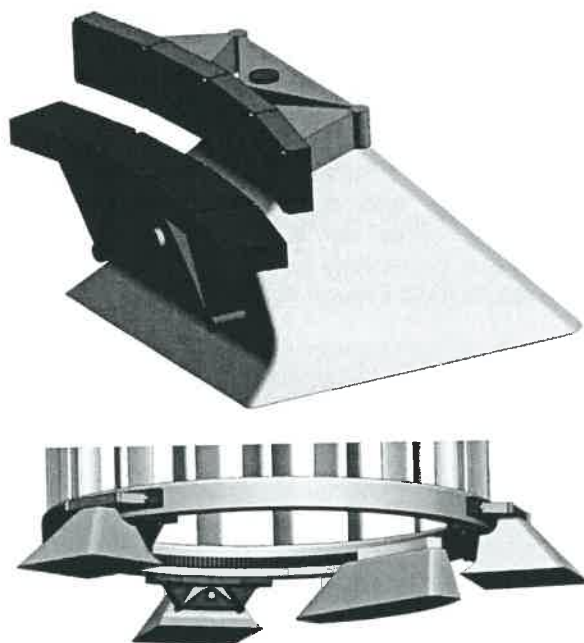
Because the turbine is perimeter supported at 3 points with hydrostatic bearing technology there are no oil based systems on board and a whole farm worth of turbines could be sunk by remote control in the face a hurricane to safely ride out the storm below the surface. The turbine could be re-commissioned with little if any direct intervention. These two points dramatically reduce risk and so cost of insurance and financing. This is a transformational and disruptive technology for the wind industry which solves multiple Energy Policy Issues.



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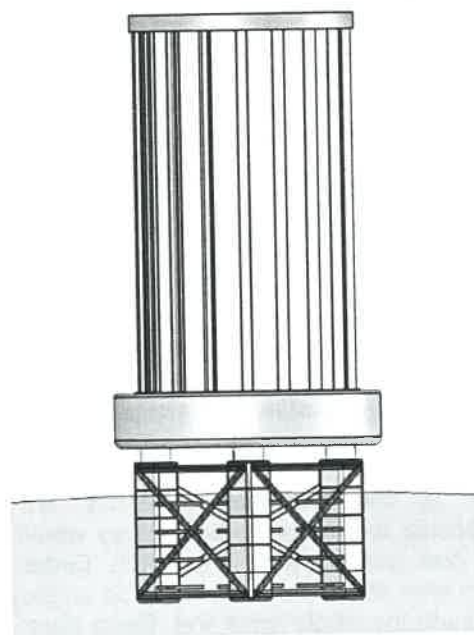
Bearing technology with the capacity and speed to support a large swept area at the perimeter has not previously been available. But now New Way Air Bearings is offering externally pressurized sea water based bearings. The guide bearings for hydroelectric turbines often use water based bearings to guide their rotors. It is recognized that such a rotor in a strong breeze will distort, not remaining flat or round. To cope with these distortions a 3 point axial support is provided. Four 1 meter wide by 2 meter long bearings are flexure mounted and ganged together on a whiffle tree at each of the three supports. The radial bearings mount in a similar fashion and cope with the change in the radius from distortion by having an axial length that is a very small percentage of the circumference.



As noted before the rotating aerodynamic structure is constructed with tension and compression principles. The three point float system is also constructed using tension and compression principles in the rotor structure is coupled to it kinematically. In this way the float system can be made exceptionally light when compared to a float system that must support a large cantilevered horizontal axis wind turbine.

A method for manufacturing large rotors for VAWTs with fluidic bearing races can be achieved through the following method. Pre-rolled or bent steel is welded together to comprise the rotor. There is a center abutment

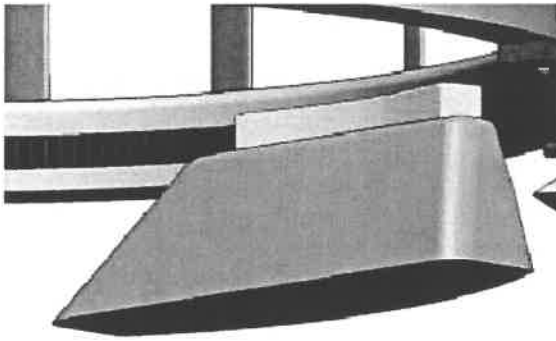
that is fit with a large spindle. The spindle may be a rolling element, plane bearing, hydrostatic or air bearing spindle for example. This spindle is used as a crane to position the steel segments of the rotor around the perimeter. The spindle also provides a way to measure the position of the steel elements before welding.



The steel segments are welded together. The center spindle is designed so that using multiple arms, it may support the entire steel rotor. The center spindle is fitted with a motor for driving the rotor. One of the perimeter pedestals is fit with modular machine tool slides. Spindles and tools may be mounted to the slides in order to precision machine the rotor in-place. After the machining operation, these slides, or another set of slides, are fit with a flame spray apparatus. The flame spray apparatus is used to deposit a coating of nickel, or other appropriate noncorrosive material, onto the prepared and machined bearing surfaces on the rotor. This coating is built up to be more than a millimeter thick. The flame sprayed nickel is then precision machined or ground. In this way a noncorrosive bearing surface with the appropriate precision can be created in the assembly yard. This avoids the problems associated with having to transport such a large rotor. Fluidic bearings are then used to kinematically support the rotor while it is spinning as a wind turbine.

HVDC PERIMETER GENERATION

Floating turbines allow for wind farms to be sited further from shore, in deeper waters and for lower installation costs. But, as discussed previously, a floating, perimeter designed VAWT can lower construction and installation costs even further. Once installed, the perimeter designed floating VAWT could be integrated with traditional magnetic induction generators, but, by combining the perimeter VAWT with an emergent electric field technology such as a high voltage DC direct drive generator, offshore wind farm costs can be lowered still further..

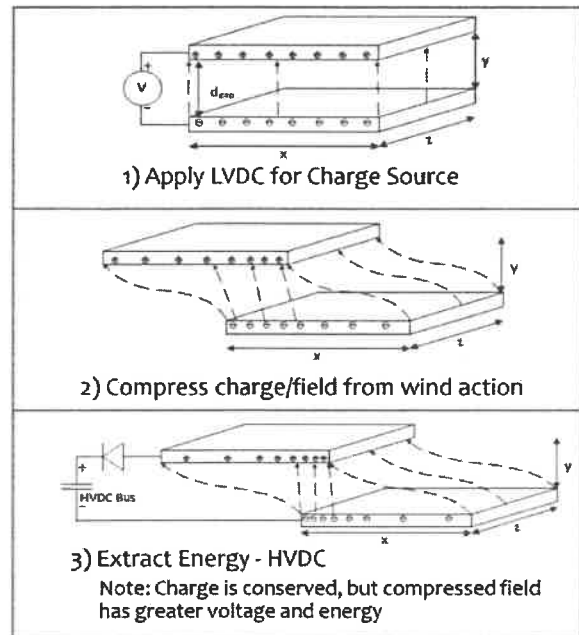


ELECTRIC FIELD TECHNOLOGY

EFT uses the inherent electric field of an electron, rather than an induced magnetic field, and, because the EFT machines as developed by EFM are based on forces from stored or static charges, they are capacitive. As a generator, EFT technology permits the production of high voltage, low current DC, rather than low voltage, high AC current as is the case for magnetic machines. For this reason, EFT eliminates steel laminations, copper coils, permanent magnets and REEs. Also, because the electric field is inherent, rather than an induced field, the di/dt component of magnetic induction power production is also eliminated, which means the EFT generator is able to produce power at any shaft speed that is greater than zero. For wind turbines, this means the efficiency of an EFT generator increases at slow speeds, because current (I^2R losses) are lower at slow speeds.

An EFT generator produces power in three primary steps:

1. **Charge:** Apply a low voltage DC (LVDC) source to aligned conductive surfaces (poles) of a phase. Due to the capacitance between the aligned conductive surfaces, charge will flow from the LVDC source to the conductive surfaces to equalize the field.
2. **Compression:** Due to wind action from the turbine blades move the charged conductive surfaces out of alignment. This movement out of alignment occurs, forces the isolated charge into an ever smaller area which increases the stored energy and voltage between the plates.
3. **Discharge:** Once the voltage between the conductive surfaces has reached a predetermined value connect the conductive surfaces to a HVDC bus.



Development of experimental facilities for investigations of microscopic mapping of fluid velocities

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INTRODUCTION

The objective of this research is to develop an instrument for tracking particles suspended in fluid using particle imaging velocimetry (PIV) technique. The main outcome of this research is to design and build a facility that is able to track individual particles in the fluid optically so that they can be analyzed computationally, the computational part of this work is not addressed here.

The applications include sorting particles in fluids based on their geometry, size and/or density, separating cells in biological sample based on their cell type [1], or the study of abrasive particle circulation in polishing [2-5].

EXPERIMENTAL PROCESS

Figure 1 shows the general setup of the PIV experiment. The fluid is placed onto an adjustable platform to set the height of the oscillating rod relative to the bottom of the transparent container. Imaging of the particles is achieved using a long range objective, high magnification camera system. Both triggering of the camera and collection of image frames is controlled by a real-time system using LabView programming. For imaging of induced streaming flows, the particles must be of near neutral density and sufficiently small to follow the flow of the fluid (particles of 10 μm to 150 diameter with density of $1.01 \text{ g}\cdot\text{cm}^{-3}$ are used for most of the experiments). Actuators and fixtures for illumination sources are fastened onto a rigid bridge frame that permits a range of positional orientations. Because of the small depth of field of the imaging system, two lasers are used to identify the focal plane during camera adjustment.

Shown in Figure 2 is a photograph of the experimental facility for the particle imaging

studies. The three legged bridge-type frame is constructed from aluminum and serves as a rigid mount for the oscillator drive mechanism, shown in Figure 3 that attaches to the underside of the frame's top.

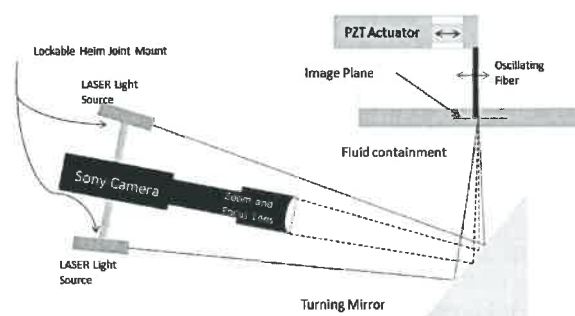


FIGURE 1. Current PIV experimental apparatus.

Experiments require the oscillating rod to be submerged within a fluid containing microspheres with depth control. This is accomplished with an adjustable height platform actuated by a micrometer.

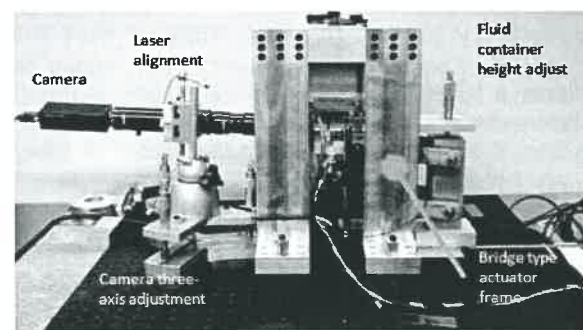


FIGURE 2. Photograph of PIV experimental apparatus.

Oscillation mechanism

A cantilevered rod is oscillated by a high bandwidth piezoelectric actuator (up to 5 kHz).

In most experiments, the amplitude of oscillation is increased by operating near to a resonant frequency. The n^{th} natural frequency can be obtained using Equation 1 where m is the mass of the beam.

$$f_n = (\alpha_n l)^2 \frac{1}{2\pi} \sqrt{\frac{EI}{mL^3}} \quad (1)$$

To obtain efficient energy transfer to the fibers attached to the piezoelectric actuator it is necessary to drive them at the resonant frequency. A molybdenum wire of 0.5 mm diameter ($E = 329$ GPa) has been used as the oscillating fiber with first mode frequency around 270 Hz.

To generate the drive signal at a defined frequency, a Direct Digital Synthesizer (DDS) chip programmed with a microprocessor through Serial Peripheral Interface (SPI) is used. Using an AD9833, resolution of 0.1 HZ is achieved for frequencies ranging from 1 milliHz up to than 2 MHz when using a 10 MHz lock. The flexure and mounting platform are made out of aluminum. Figure 3 shows the fiber actuator mechanism.

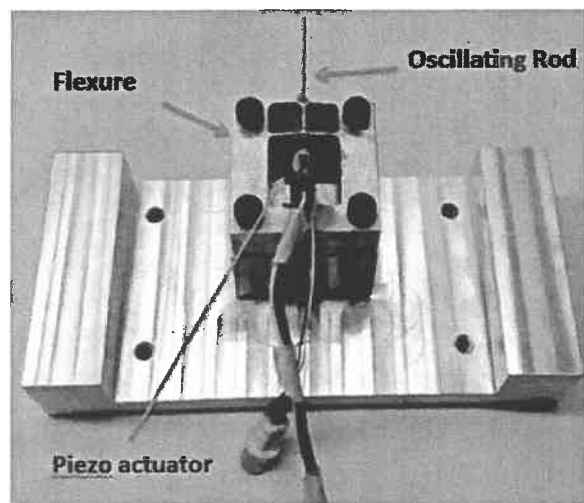


FIGURE 3. Oscillator drive mechanism showing the oscillating rod that has its axis perpendicular to the motion of the piezo electrically actuated flexure stage.

A block diagram of the system used to drive the PZT actuator to oscillate the rod in the experiments is shown in Figure 4. Information regarding the signal type and frequency are transmitted from Labview to a microprocessor. The class A power driver amplifies the DDS signal up to 150 volts to drive the PZT actuator.

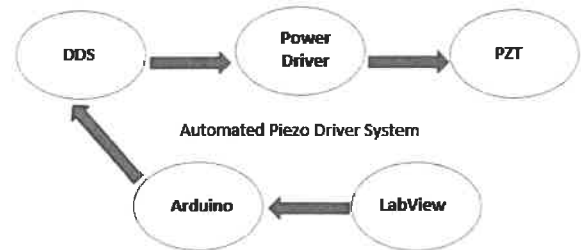


FIGURE 4. Automated piezoelectric driver system.

Camera positioning stage

Previously the camera manipulation to view particles at different heights in the fluid was carried out manually which made it difficult to find and adjust the focal plane at the high magnification of this system. Figure 5 shows the apparatus for the automatic positioning of the camera which is also microprocessor controlled. Three stepper motors are attached to a platform with 120 degree pitch and the combination of stepper movements provides pitch, roll and height adjustments. A universal vise is also attached to the top of this platform and is used for initial coarse alignment of the camera. Two joysticks (two-axis movement for each) are linked to the microprocessor to control the stepper motors. The analog joysticks provide direction and variable speed control of the stepper motors.

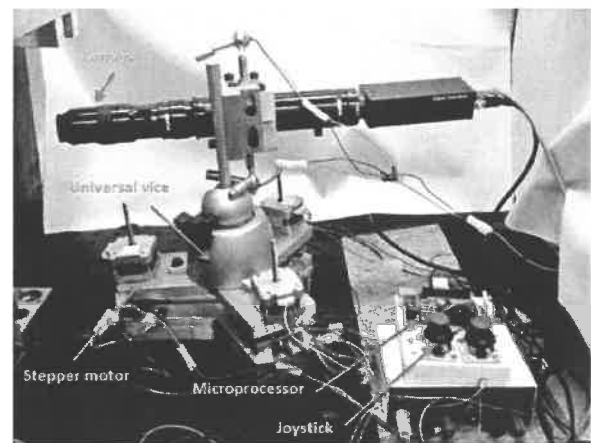


FIGURE 5. Camera positioning stage with microprocessor control.

Illumination

Illumination of particles plays an important role in recording the high quality movies with the camera. Illumination uses combinations of 40W RGBW LED high brightness sources. Figure 6

shows the circuit for the LED brightness adjustment. Work is in progress to add parabolic mirrors to the LED's to direct light and to implement a platform for adjusting light in orientations that optimize scattering of light from particle without stray reflection from other components being transmitted to the camera.

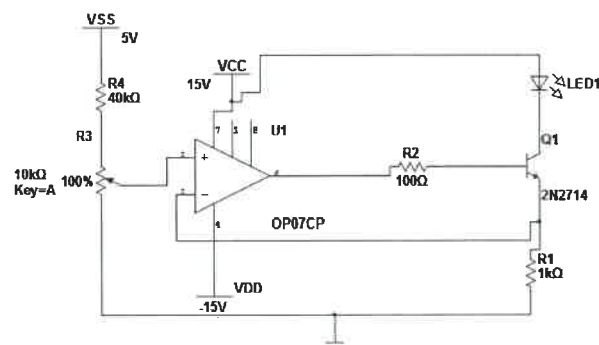


FIGURE 6. LED brightness adjustment circuit

A right-angle prism, or 90 degree, mirror with mirror-coated surfaces is attached on an adjustable leg attached to a height adjustment stage as shown in Figure 7. The LED is attached to the same stage and shines on one side of the prism mirror and reflects towards the bottom of the rod. Images are captured by the camera while viewing via the mirror on the opposite side of the prism. This method enables only the light scattered back from the flow zone (the rod and particles in the container) to be captured. To further reduce stray light, the entire setup surrounding the flow region is painted a matte (chalkboard) black.

As shown in Figure 7, the LED is attached on an inclined plate facing towards the mirror. To direct as much light as possible to the flow zone, the optimum angle was calculated for the LED which is shown as a in Figure 7. The adjustable stage leg is utilized because the distance, h , between the rod and prism is critical for the image of the rod to appear on the center of the mirror face. By placing the reflection of the rod on the center, the flow zone area captured by the camera can be maximized through orientation of the camera with respect to the prism. The optimal angle, a , for LED illumination can be calculated using Equation 2, 3, and 4.

$$b = \tan^{-1} \left(\frac{4h}{x} \right) - \frac{\pi}{4} \quad (2)$$

$$c = \frac{\pi}{2} - b \quad (3)$$

$$a = \frac{\pi}{4} + b \quad (4)$$

By substituting the dimensions of the setup with these parameters, angle a is obtained as approximately 68 degrees.

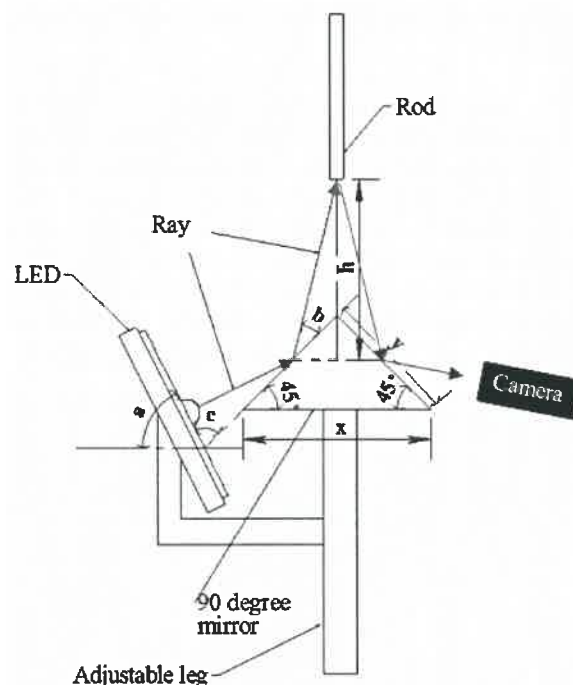


FIGURE 7. Diagram of the illumination system. The blue line indicates LED illumination and red is the reflected image to the camera.

RESULTS

Images from a recorded video of the stationary and oscillating rod are shown in Figure 8. No liquid solution was used during this experiment. The bottom images were taken after grinding the tip of the rod flat. As seen from the images, flattening the rod's tip improved the view of the rod. When oscillated with a 270 Hz signal applied to the piezoelectric, the rod moved in an elliptical path.

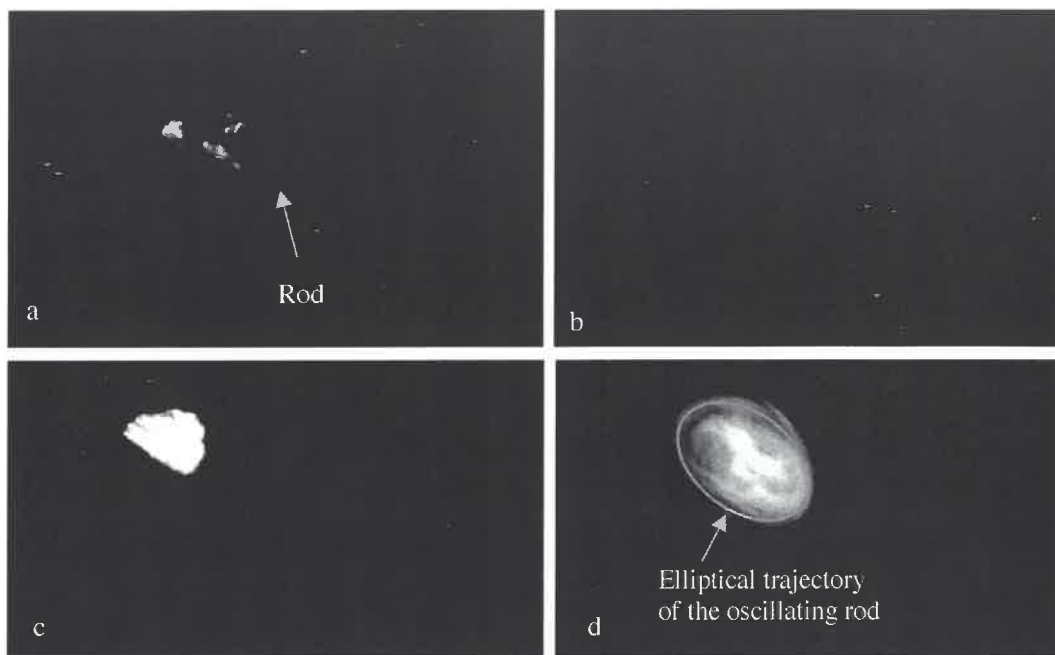


FIGURE 8. Images of the original rod end when a) stationary and b) oscillating, and images of the flattened rod end when c) stationary and d) oscillating.

Figure 9 shows an image with the rod oscillating in distilled water with microsphere particles with concentration of 42600 particles per 400 mL and diameter of 45-53 μm . Generally, movies obtained with this system showed particles close to the rod circulating either as vortices adjacent to the rod or around the oscillating rod itself. These circulating patterns were strongly dependent on the boundary conditions of this experiment. There are some particles precipitated on the glass slide that are stationary, as well as bright spots which need to be removed using digital filters.

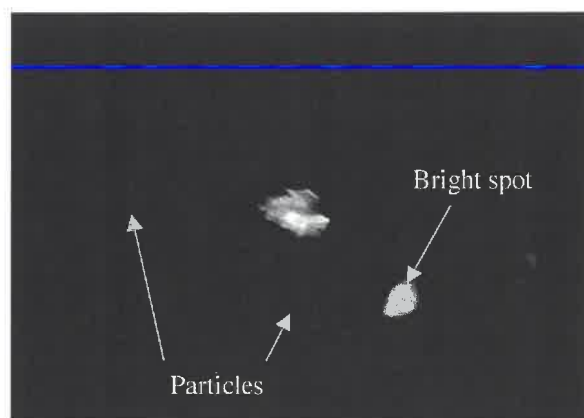


FIGURE 9. Image showing moving particles and a bright spot.

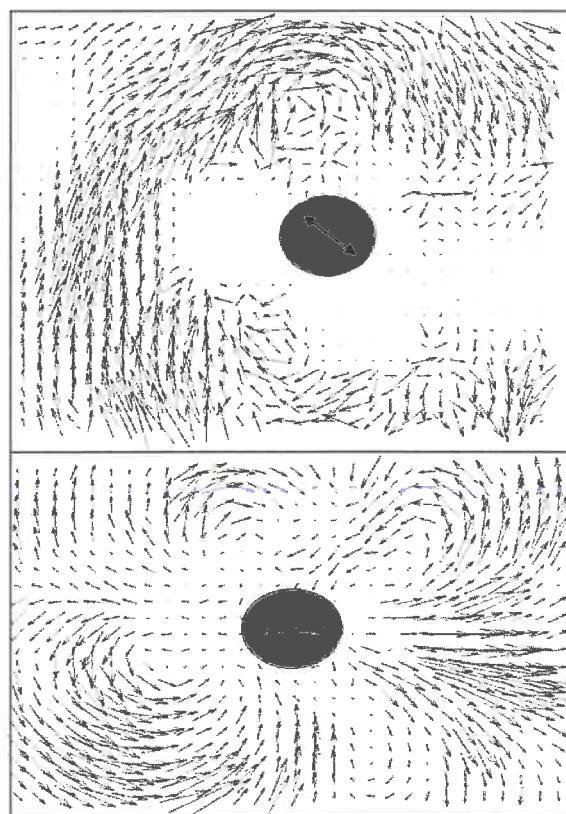


FIGURE 10. Early PIV software vector data. Arrows inside the ellipses indicates the rod's oscillation direction.

Figure 10 depicts initial PIV results from images obtained from a single-probe flow experiment for which one movie frame is shown in Figure 9. The particle velocity field is reconstructed from video frames using commercial particle image velocimetry software, Dynamic Studio, from Dantec Dynamics. Based on theoretical studies of two dimensional flows [6] it is observed that in single probe flow experiments particles will move in a trajectory forming four symmetric streaming cells around the rod. In practice, the rod is of finite length and flows are expected to depart from this ideal model in the regions near to the surface and around the free end of the rod. A four vortex pattern (often called a chevron pattern) is visible in the lower PIV velocity map in Figure 10. However, as shown in upper image of Figure 10, some experimental results shows particles following different paths. These experiments have different rods, particle concentrations and size, and the rods are located within a circular reservoir without precise positioning. All parameters in these early experiments are now being more precisely and accurately measured.

CONCLUSION

Several improvement to the experimental facility of particle manipulation project have been added. Quality of recorded videos has increased by adopting a new lightening system. Finding the focal point and manipulating the camera to capture different flow regions has been made more convenient by building a motorized 3-axis motion control stage. Changing the frequency of the rod is handled by a custom developed high resolution signal generator.

FUTURE WORK

Improvement to image quality will continue by modifying the physical setup. Once the system configuration is optimized for raw image quality, images will be processed using digital filters. One filter for instance is needed for removing particles attached to the surface of the glass slide. After filtering, particle image velocimetry (PIV) studies will be carried out and compared to theoretical models. Finally, a second rod will be added to the system for analysis.

Studies of the particle flow patterns and comparison of these to theoretical models [6] are in early stages. A more systematic study to assess the influence of boundary conditions, Reynold number dependence, and multiple oscillators flow patterns are planned.

ACKNOWLEDGEMENT

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Estimation of Heat Generation in a Linear Motion Bearing

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INSTRUCTIONS

Linear motion bearings are used as a feeding unit in machine tools, LCD and semiconductor systems because of high stiffness and accuracy. And they are easy to use compared with other bearings. When the linear motion bearings are used as a feeding unit for above applications, very high speed is required for high throughput. However, this makes a large amount of heat caused by friction force of the bearings, which obviously causes thermal deformation and deteriorates motion accuracy. Therefore, it is important to consider effect of heat in designing a linear motion bearing stage.

In order to predict heat generation in the linear motion bearings, we used friction force obtained while the bearings are moving. The friction force of the linear motion bearings is affected by several parameters such as external load, lubrication, preload, velocity as well as bearing type and size. So the friction measurement system of the linear motion bearings is designed considering these parameters. Then heat flow rate is predicted from the measured friction force, and temperature rise is estimated from the heat flow rate and the FEM tools. Finally, the predicted temperature rise is compared with actual measurement.

FRICTION MEASUREMENT SYSTEM

Figure 1 shows a flowchart for heat prediction. First, friction of the linear motion bearings is measured using a specially designed friction measurement system under several conditions. Then, heat flow rate is calculated under assumption that a work done by the friction is converted into the heat. Then temperature distributions using FEM analysis are compared with actual measurements by thermocouples.

Figure 2 shows the friction measurement system designed in this paper. It consists of two opposing linear motion guides and surrounding boxed structure for applying external load, and independent feeding unit for moving the bearings. In order to apply the external load on the bearings, two linear motion rails and bearing blocks are oppositely attached, and surrounding

box typed structure is installed on the bearing blocks. The screw on the box typed structure is adjusted to apply load, and its magnitude is measured using a loadcell which is installed between surrounding box and upper bearing block. A feeding unit is installed in parallel with the two opposing linear motion guides. So, the linear motion bearings can move continuously. Another loadcell is installed between the feeding unit and surrounding box to measure the friction force of the bearings in an axial direction. The used loadcell can measure compression and tension force. Therefore the friction force is continuously measured in both directions while the bearings are moving reciprocally.

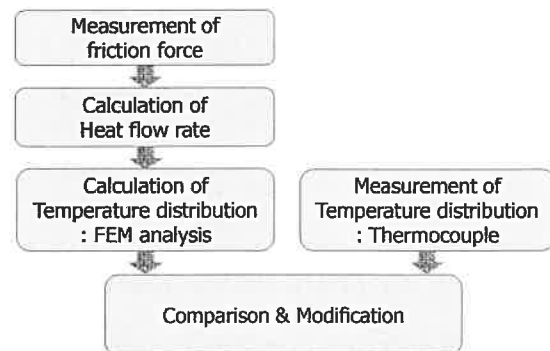


FIGURE 1. Flowchart for heat prediction

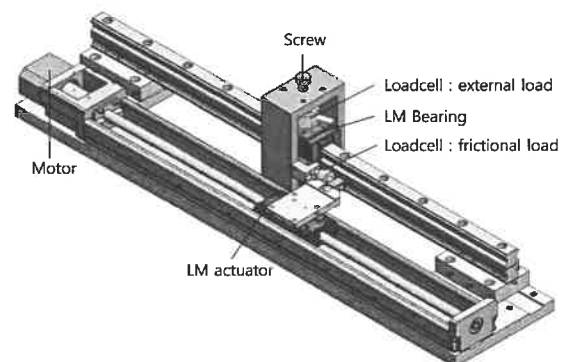


FIGURE 2. Friction measurement system

FRICITION MEASUREMENT RESULT

Friction characteristics of a linear motion bearing (THK HSR30R) are evaluated under several different conditions (velocity, external load, preload, bearing size). Figure 3 (a) and (b) show friction force measured with external force and velocity, respectively. The friction force increased with external force and velocity. And relationship between friction force and velocity presents typical Stribeck curve. However, we have no interest in the friction force in small velocity less than 50 mm/s because heat generation is very small in the region.

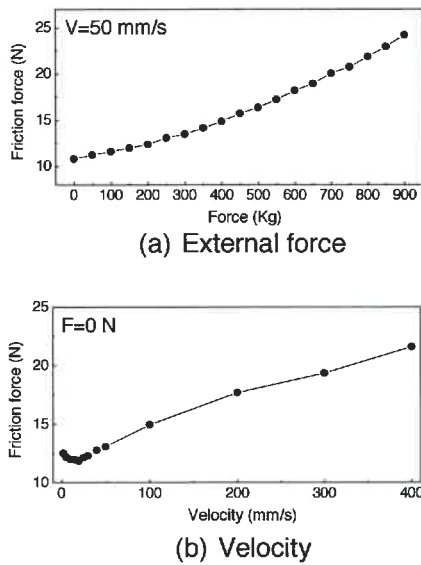


FIGURE 3. Friction force

The friction force is generalized using the Box-Behnken method with external force, velocity, and preload of the bearing, which is expressed by Eq. (1).

$$\begin{aligned}
 F_{friction} &= Av_f + BF_n + CP_b + A'v_f^2 + B'F_n^2 + C'P_b^2 \\
 &+ A''v_f F_n + B''v_f P_b + C''F_n P_b + D \\
 &= 21.6v_f + 5.5 \times 10^{-4} F_n - 4.6P_b + 5v_f^2 + 4.3 \times 10^{-8} F_n^2 \\
 &+ 5.6P_b^2 + 4.9v_f P_b + 5.3 \times 10^{-5} F_n P_b + 6.4
 \end{aligned} \quad (1)$$

Where, v_f is velocity of the bearing, F_n is external force, and P_b is preload of the bearing. Figure 4 shows friction force obtained while the linear motion bearing is moving for a long time. Most importantly, the friction force is constant regardless of movement time of the bearing. Positive value in the friction force means compression of the loadcell.

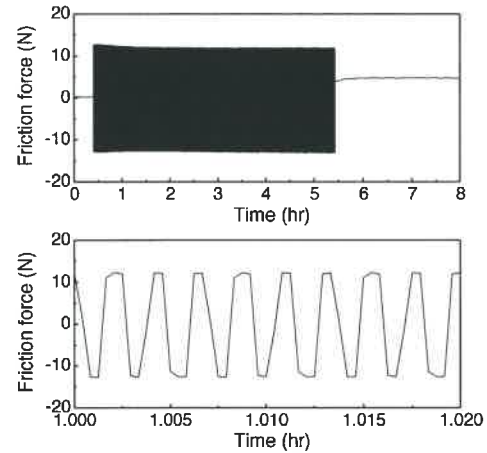


FIGURE 4. Friction force with time

Figure 5 shows generalized friction force using the Eq. (1) and measured data which are not used for the equation. The measured data showed a good agreement with the generalized friction force equation.

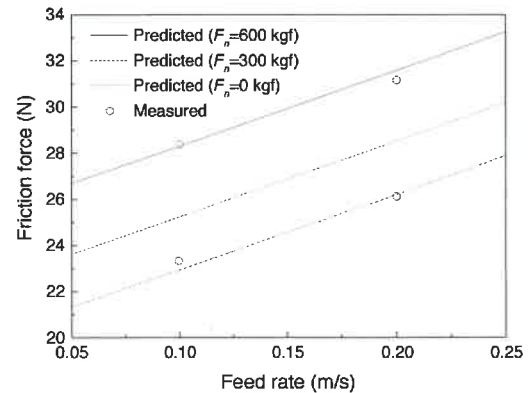


FIGURE 5. Generalized friction force equation

TEMPERATURE MEASUREMENT AND COMPARISON

It is assumed that the friction force is transferred into heat. So heat flow rate, Q , generated while a linear motion bearing is moving is predicted from the measured friction force as Eq. (2).

$$Q = F_{friction} \cdot v_f \quad (2)$$

From the predicted heat flow rate, temperature distributions and thermal deformations of the linear motion bearings are theoretically estimated using the FEM analysis. We compared estimated temperatures with measured temperatures in order to verify the estimated heat flow rate. Figure 6 shows temperature distributions obtained from the

predicted heat flow rate. The heat flow rate is given in contact areas between the rail/block grooves and balls. The boundary conditions are conduction inside bearings and rails, and convection between surfaces of the bearings and atmosphere.

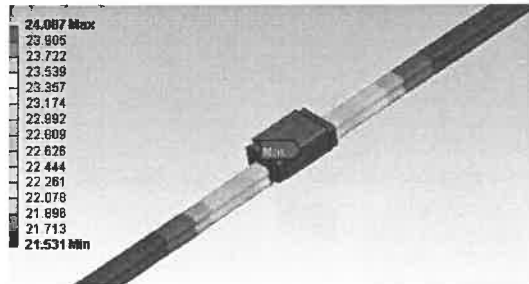


FIGURE 6. Temperature prediction

Figure 7 shows temperature measurement positions. The temperatures are measured using thermocouples in rails and bearing blocks while the linear motion bearings are moving during 5 hours. The measured temperatures are shown in Figure 8. The predicted temperatures are well agreed with measurements by comparing Figure 6 and 8. In conclusion, heat generation during movement of linear motion bearings can be predicted from the proposed friction measurement system.

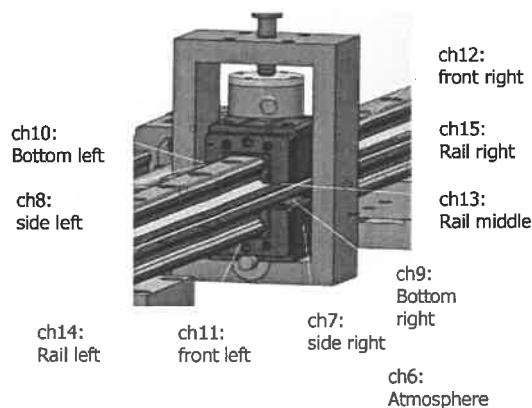


FIGURE 7. Temperature measurement positions

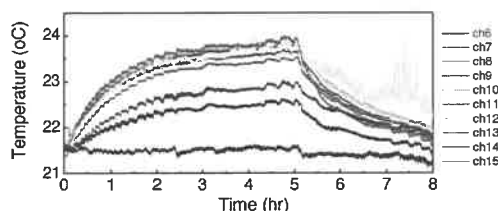


FIGURE 8. Temperature measurement

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Design of Compact Sensing System of VCM Spherical Actuator

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INTRODUCTION

Recently, the study of spherical actuators to perform multi degree of freedom movements has increased. The spherical actuator is an electric machine that is able to tilt its shaft on multiple degrees of freedom using electromagnetic force. Various types of spherical motors have been developed. Lee et al.[1-4] devised variable reluctance spherical actuators. Yan et al.[5,6] developed a spherical actuator similar to Lee's; their drive principles are actually synchronous.

The existing spherical actuators have sensing and guide mechanism is outside of actuators. It causes bulk size and unpractical for application. This means that new type of sensing system placed inner side of actuator is necessary for miniaturization of the spherical actuators.

For this reasons, we propose a new type of sensing system for spherical actuator to miniaturize total system. After designing, we verify the performance of proposed sensing system by comparing with rotary encoder.

CONCEPT OF THE VCM SPHERICAL ACTUATOR

Figure 1 shows conceptual design of VCM (Voice Coil Motor) spherical actuator. This spherical actuator uses VCM based on the Lorentz force principle. It is composed of a stator and a moving coil type rotor.

The stator of the actuator has eight permanent magnets and steel yoke for high uniform magnetic flux density. Yoke is divided two parts: outer yoke and inner yoke. The yokes create closed magnetic flux loop. The permanent magnets are attached to outer yoke.

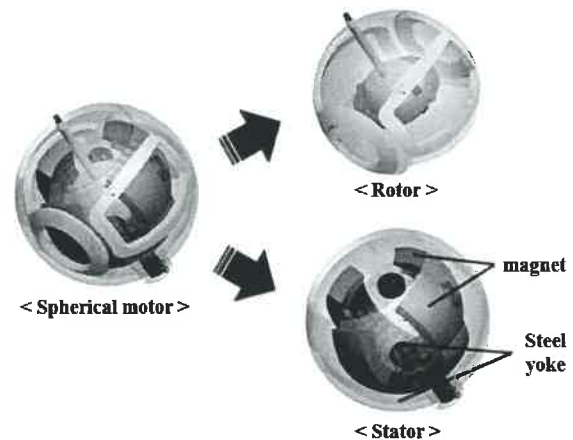


FIGURE 1. Concept of the spherical actuator

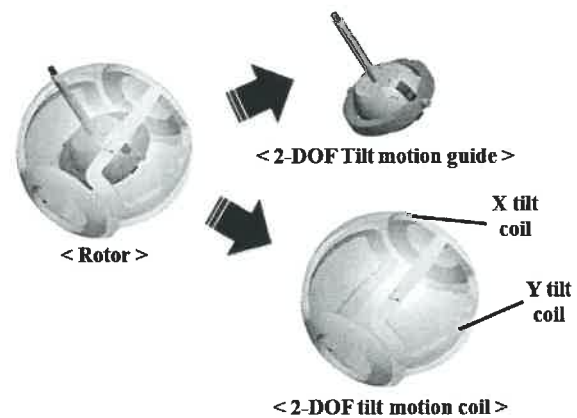


FIGURE 2. Rotor and Guide system

The rotor and guide system is shown in Figure 2. The rotor is composed of a two degree of freedom gimbal guide mechanism and four coils to drive the tilt motions. There are two pairs of different shaped coils are used to drive the X and Y-axis tilt motions. X-coil is made circular shape in order to getting independence from rotating motion. Therefore, spherical actuator can produce constant torque when it rotates on X-axis. It is also independent of Y-axis rotation.

For this reason, circular coils generate the X-axis tilt torque and rectangular coils generate the Y'-axis tilt torque independently. These coils are moved with the gimbal guide mechanism. The coil jig is fixed on the shaft of the gimbal guide mechanism.

SENSING SYSTEM OF THE VCM SPHERICAL ACTUATOR

The compact sensing system is important for miniaturization of the VCM Spherical actuator. To achieve high resolution and compact size, the proposed sensing system is composed of a gimbal guide system, permanent magnet arrays and hall sensor.

Sensing mechanism

Figure 3 shows the proposed sensing system. Following the rotation of guide system, sensor system measures each axis tilt motion independently.

As shown in Figure 3, two magnet arrays with 90° phase difference are fixed on the inner yoke and hall sensors are placed on the X-axis tilt guide for sensing X-axis tilt motion. In the same way, Y' magnet arrays are fixed on the Y'-axis tilt guide and hall sensors are placed on the X-axis tilt guide.

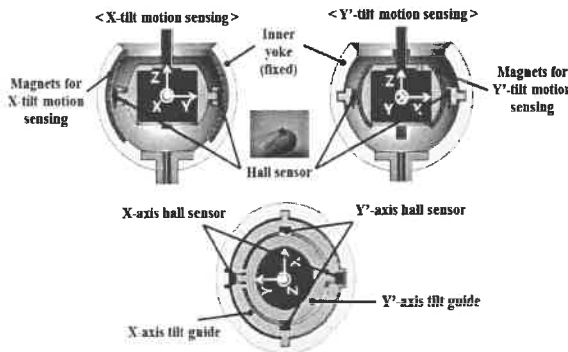


FIGURE 3. The sensing system of the spherical actuator

Figure 4 shows the expected signals of two hall sensors at one axis. As tilting with X-axis or Y'-axis, two sinusoidal signals with 90° phase difference is measured. The tilting angle is calculated based on the conversion algorithm [7]. The conversion algorithm is shown in Table 1.

After conversion processing, the expected angle is shown in Figure 5.

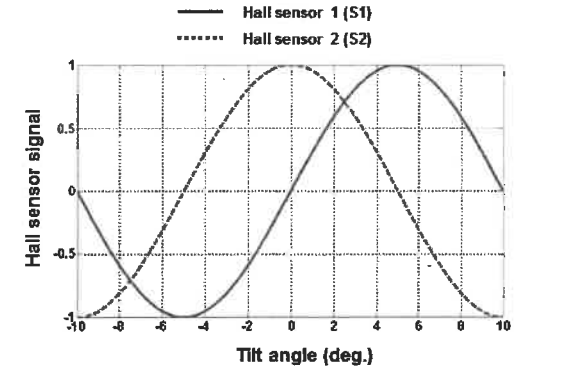


FIGURE 4. The expected sensor signal of the proposed sensing system

TABLE 1. The degree conversion algorithm

Range (deg.)	S1 > 0	S2 > 0	S1 > S2	Formula for tilt angle
-10 ~ -7.5	X	X	X	$-10 + \tan^{-1}(S1/S2)$
-7.5 ~ -5	X	X	O	$-5 - \tan^{-1}(S2/S1)$
-5 ~ -2.5	X	O	O	$-5 - \tan^{-1}(S2/S1)$
-2.5 ~ 0	X	O	X	$\tan^{-1}(S1/S2)$
0 ~ 2.5	O	O	X	$\tan^{-1}(S1/S2)$
2.5 ~ 5	O	O	O	$5 - \tan^{-1}(S2/S1)$
5 ~ 7.5	O	X	O	$5 - \tan^{-1}(S2/S1)$
7.5 ~ 10	O	X	X	$10 - \tan^{-1}(S1/S2)$

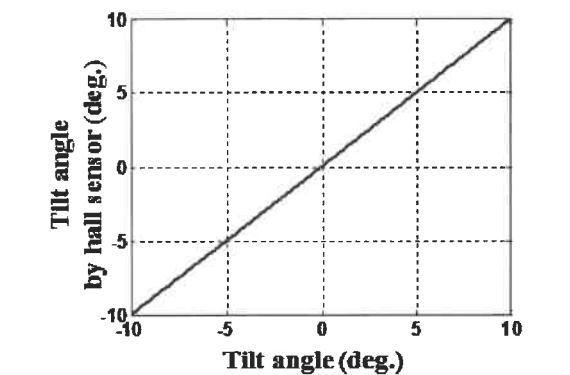


FIGURE 5. The expected angle data after sensor signal conversion processing

Verification of performance

After designing the proposed sensing system, the verification of performance is necessary for replacing outside sensor. In the verification experiment, the rotary encoder is used for reference sensor. By comparing with reference data, the possibility of using the proposed sensing system is expected.

The experiment system for verification is composed of outside rotary encoder and the proposed sensing system with guide system in the spherical actuator.

Following the Y' gimbal guide, Y' tilting angle is measured by outside rotary encoder and proposed sensing system. The data measured by the proposed sensing system is converted into angle data. After signal processing, a comparison of the proposed sensing system angle data with rotary encoder angle data is represented in Figure 6. The difference between tilt angle measured by hall sensor and reference is represented in Figure 7.

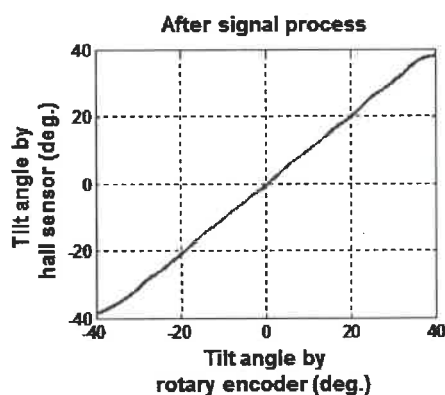


FIGURE 6. The Y'-axis tilt angle measured by rotary encoder and proposed sensing system

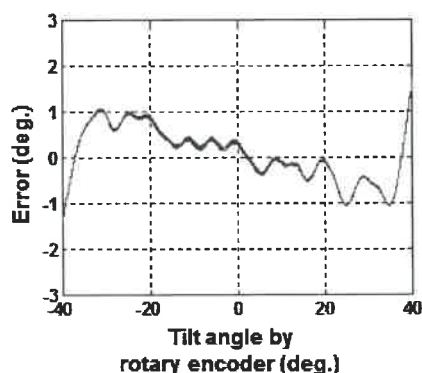


FIGURE 7. The difference of Y'-axis tilt angle measured by rotary encoder and proposed sensing system

The X-axis measurement is independent of Y'-axis measurement. The experimental system for X-axis measurement permits only X-axis tilt motion. X-axis tilt angle is measured by outside

rotary encoder and proposed sensing system. In the same way, X-axis hall sensor data is converted into angle data and a comparison of the proposed sensing system angle data with rotary encoder angle data is shown in Figure 8.

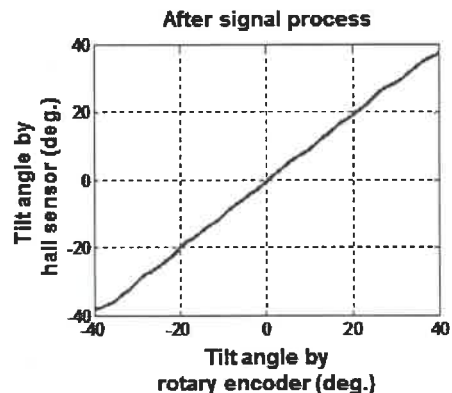


FIGURE 8. The X-axis tilt angle measured by rotary encoder and proposed sensing system

Similarly, X-axis tilt angle measured by rotary encoder and hall sensor is different. The error of X-axis tilt angle is represented in Figure 9.

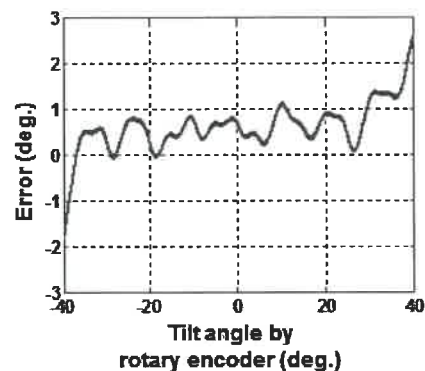


FIGURE 9. The difference of X-axis tilt angle measured by rotary encoder and proposed sensing system

The proposed sensing system with guide mechanism can place the inside of VCM spherical actuator. Therefore, spherical actuator maintains compact size and does not add any outside sensor. In the actual sensing system, however, measured angle data is not same with rotary encoder. So the calibration of measured data is necessary.

CONCLUSION

In this study, a new type of sensing system using hall sensor is proposed and verified performance. As the difference between the proposed sensing system and reference rotary encoder exists, the proposed sensing system needs calibration for replacing commercial encoder. The proposed sensing system is used for compact multi degree of freedom spherical actuator. The proposed sensing system can be installed in the spherical actuator and replace rotary encoder which is set up the outside of spherical actuator.

Future work includes calibration of the sensing system and verification performances of the spherical actuator using this proposed sensing system.

ACKNOWLEDGEMENT

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EFFECTS OF PRESSURE PULSATIONS ON AN OIL HYDROSTATIC SPINDLE

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ABSTRACT

Hydrostatic spindles and slides exhibit errors due to the ever-present pressure ripple of pumps. Design and testing of a high pressure (4 MPa) oil supply is described that results in nanometer-level spindle error. Hydraulic design guidelines derived by Viersma are experimentally verified using static and dynamic pressure ripple measurements. The root total of the Cumulative Power Spectrum (CPS) is used to verify hydraulic system performance enhancements. Best results were obtained using a suppressor and regulator close to the pump.

BACKGROUND

Positive displacement pumps are used to supply high pressure oil to hydrostatic bearings. Compared to other positive displacement pumps, fixed displacement gear pumps are the most commonly used due to their high reliability and low cost [1]. These pumps exhibit flow fluctuations, or flow ripple, that induce pressure variation in the bearing films causing errors. The pressure pulses occur at the pump fundamental frequency and harmonics as well as the driving gear tooth pass frequency and harmonics. The harmonics can be reduced, but not eliminated, by increasing the number of teeth on the driving gear [1].

The most comprehensive and deterministic approach to suppression of pulsations in hydraulic systems is provided by Viersma. In his book, he derives transfer functions for each element of the hydraulic supply to arrive at the following design guidelines [2]:

- suppressor (or accumulator) placed close to the pump acts as a filter for high frequencies
- pressure control valve (or regulator) close to the pump controls the low frequency pressure
- combination of suppressor and regulator close to the pump provides an effective wide band filter
- selective filters can be added to further reduce specific frequency disturbances

HYDROSTATIC SPINDLE

The hydrostatic spindle under test uses step-compensation rather than restrictors with pockets. The rotor shown in Figure 1 uses conical bearing surfaces to provide axial, radial, and tilt load capacity. The high structural stiffness of the rotor contributes to static stiffness while thin oil films result in excellent damping [3].

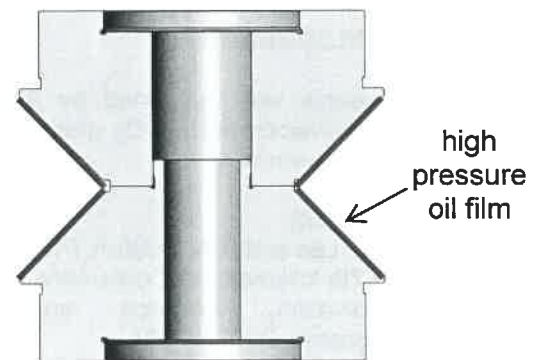


FIGURE 1. A cross-sectional view of the bi-conic rotor. The compact and stiff rotor permits the use of high pressure oil.

Stiffness and load capacity with 4 MPa (600 psi) oil are summarized in Table 1. Since the rotor is stiff in the direction of the force vectors of the film, higher pressure oil provides higher stiffness and higher load capacity than a more compliant structure [4].

TABLE 1. Measurements of stiffness and load capacity of an oil hydrostatic 4R bi-conic spindle with oil supply pressure of 4 MPa (600 psi).

	Axial	Radial	Tilt
Stiffness	350 N/ μ m	130 N/ μ m	1.1 N•m/ μ rad
Load capacity	9,000 N	3,000 N	300 N•m

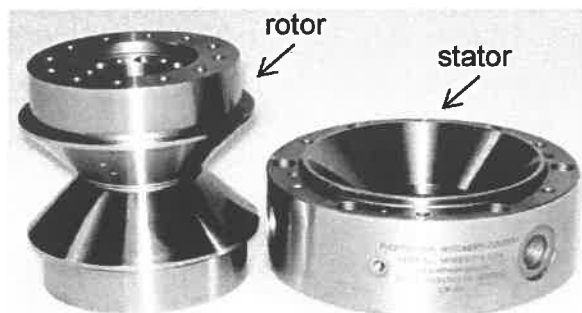


FIGURE 1. Rotor and stator of the 4R bi-conic spindle under test.

PULSATION SENSITIVITY

Theoretically, a symmetrical bearing design is insensitive to flow surges due to common mode rejection [3] but this is seldom the case in the presence of structural compliance. For example, as a pressure pulse enters the bearing films shown in Figure 1, the rotor centerline remains unchanged since both films are affected equally. However, the increased pressure causes the rotor to deflect axially and radially which results in errors due to non-constant (pressure) conditions.

To predict the effects of pressure pulsations on errors, the concept of a frequency dependent "pulsation sensitivity" is introduced. The term combines the effects of pulses on the dynamic structural compliance and fluid dynamics in the bearing film.

Finite Element Analysis (FEA) is used to model the pulsation sensitivity due to static pressure changes. The pressure profile shown in Figure 2 is applied to a rotor half. A fixed boundary condition is applied at the middle of the rotor.

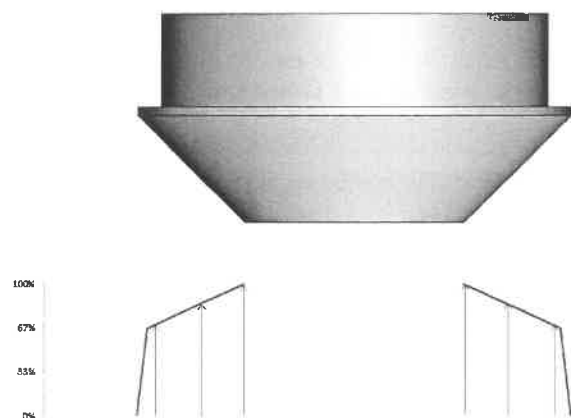


FIGURE 2. Pressure profile for step compensation applied to rotor half.

The static pressure profile applied to the bearing surfaces of the rotor causes a deflection of the rotor as shown in Figure 3. FEA predicts that at low frequencies, changes in supply pressure will result in axial errors of 0.3 nm per 1 kPa. This result was verified by increasing the supply pressure to the spindle from 2.8 MPa (400 PSI) to 4.8 MPa (700 PSI) and recording the change in flying height of the rotor with a capacitive sensor. The result is plotted in Figure 4 and the measured pulsation sensitivity for static pressure changes is 0.3 nm/kPa (the slope of the line).



FIGURE 3. The stretch of the rotor predicted by FEA due to a supply pressure of 4 MPa is 1.3 μm . The low frequency pulsation sensitivity is 0.3 nm/kPa.

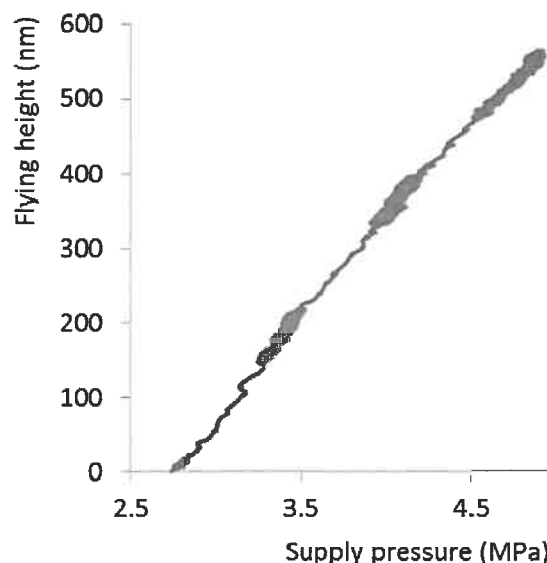


FIGURE 4. Stretch experimentally verified by measuring the flying height from 2.8-4.8 MPa (400-700 PSI). The static pressure sensitivity is 0.3 nm/kPa.

These results are useful in estimating how much asynchronous will be caused by a given level of pressure ripple. For example, to keep static-to-low-frequency asynchronous error caused by pressure ripple below 1 nm, the hydraulic pulsations should be kept below 3 kPa. This defines the target performance of the hydraulic system.

HYDRAULIC SYSTEM

A schematic of the hydraulic system under test is shown in Figure 5. The system provides 1 L/min of low viscosity spindle oil (Mobile Velocite No. 6) to the spindle at 4 MPa (600 PSI). The combination of high pressure, low viscosity, and low flow with low pumping losses is atypical of standard hydraulic power units so a purpose-built system was developed (shown in Figure 6).

The hydraulic system features a 0.7 kW brushless servo motor (Baldor BSM33) connected to an external gear pump (Concentric 1 cm³) with PID speed control to maintain output oil pressure at 4.8 MPa. This configuration provides the minimum pumping power loss [5]. A pressure relief set at 5.5 MPa is used as a safety valve to protect the system against overload. Under normal operating conditions there is no flow through the relief valve. The oil then passes through a 10 μ m particulate filter (the nominal bearing film thickness is 16 μ m). Then the oil flows through an in-line nitrogen charged noise suppressor (Wilkes and McLean WM 3056) and a high pressure oil regulator (Graco 217576) set at 4 MPa. Finally, the oil enters the spindle and flows across the bearings. Non-contact pressurized air seals prevent leakage and provide a sufficient pressure to assist drainage back to the tank without causing excessive aeration of the oil.

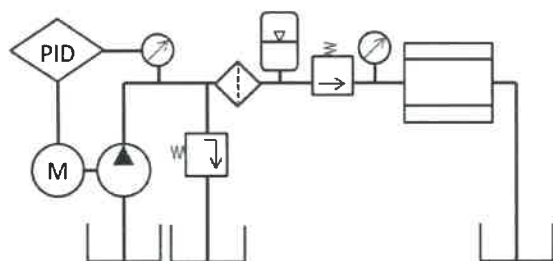


FIGURE 5. Schematic of the bespoke hydraulic power unit with minimal pumping losses and reduced pulsations.

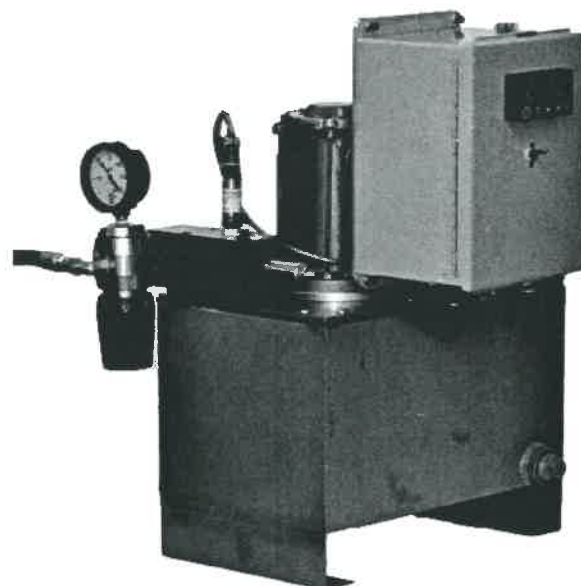


FIGURE 6. Hydraulic power unit with external gear pump and brushless servo motor. The pressure is maintained via PID speed control. The unit also features an in-line suppressor with a high pressure regulator.

RESULTS

Oil pressure and axial displacement Power Spectral Densities (PSD) are measured as a function of hydraulic system configuration. A piezoresistive pressure sensor (Omega PX429) with a bandwidth of DC to 1000 Hz measures the supply pressure at the auxiliary oil inlet as shown in Figure 7. An ultra-low noise (0.4 nm pk at 15 kHz bandwidth) capacitive sensor (Lion Precision 2G-C8-1.2) targets a testball in the axial direction of the non-rotating spindle. A dynamic signal analyzer (HP 35670A) acquires power spectral densities and the root of the Cumulative Power Spectrum (CPS) from 1 Hz to 200 Hz provides the standard deviation of the total error [6]. Errors below 1 Hz are excluded due to thermal drift which will be addressed in future work.

PSDs for the pressure and axial displacement are shown for just-the-pump (Figures 8 and 9), pump with in-line suppressor (Figures 10 and 11), and pump with suppressor and regulator (Figures 12 and 13). The quietest configuration is with the suppressor and regulator close to the pump as suggested by Viersma [2].

The most dominant peaks in the pressure spectrum of Figure 8 are the tooth pass frequency (81 Hz) and tooth pass harmonics. The pump speed (7.5 Hz) and harmonics also contribute but at much lower amplitudes. The root total CPS of 19 kPa is well above the desired level of pulsations based on the static pulsation sensitivity estimate. The displacement spectrum in Figure 9 mimics the pressure spectrum and, as predicted, the CPS error exceeds the desired level.

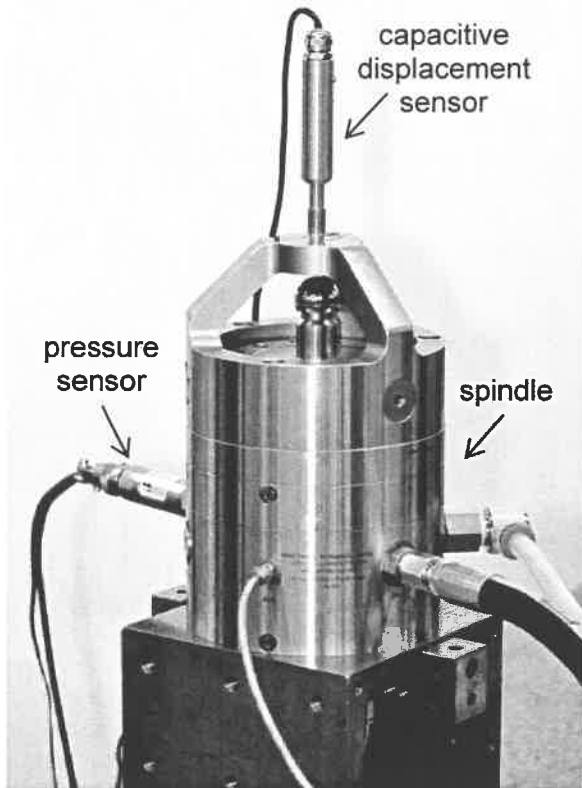


FIGURE 7. Setup showing capacitive sensor measuring axial displacements and pressure sensor measuring pressure pulsations.

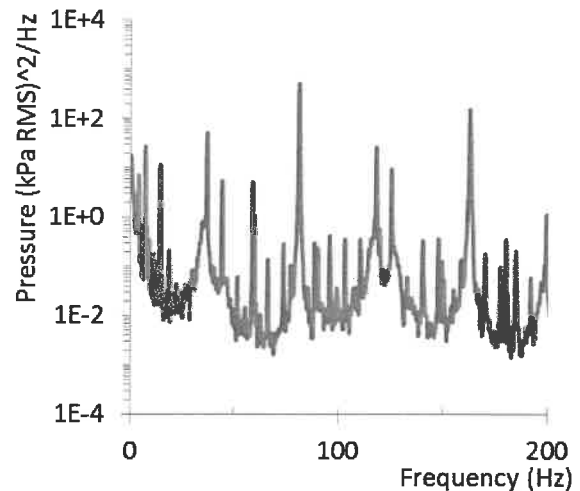


FIGURE 8. Just-the-pump pressure PSD with supply pressure at 4 MPa. Root total CPS value is 19 kPa.

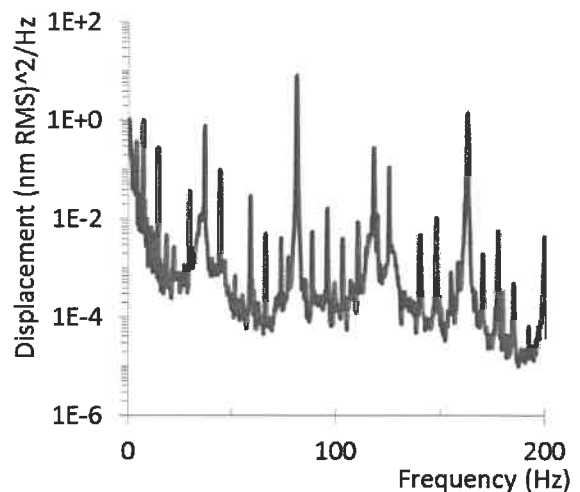


FIGURE 9. Just-the-pump displacement PSD with supply pressure at 4 MPa. Root total CPS value is 2.5 nm.

With the addition of an in-line pulsation suppressor, the spectra are significantly improved as shown in Figure 10 and Figure 11. The pulsation amplitude at the tooth pass frequency was reduced from 520 (kPa RMS)²/Hz to 13 (kPa RMS)²/Hz. The corresponding displacement at that frequency was reduced from 8 (nm RMS)²/Hz to 0.4 (nm RMS)²/Hz. The root total CPS values are 3 kPa and 0.7 nm respectively.

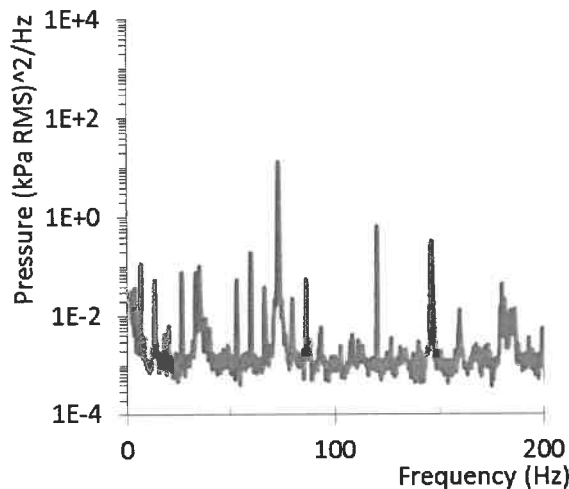


FIGURE 10. Pump and suppressor pressure PSD with supply pressure at 4 MPa. Root total CPS value is 3 kPa.

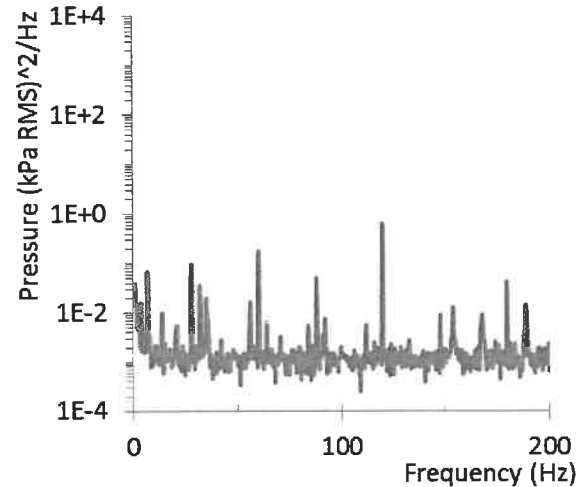


FIGURE 12. Pump with suppressor and regulator pressure PSD with supply pressure at 4 MPa. Root total CPS value is 1 kPa.

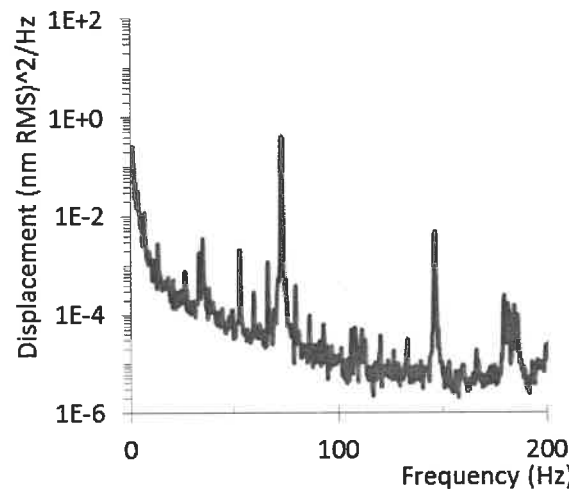


FIGURE 11. Pump and suppressor displacement PSD with supply pressure at 4 MPa. Root total CPS value is 0.7 nm.

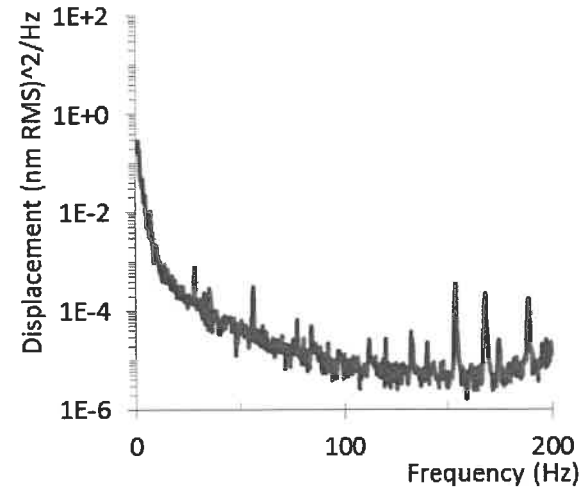


FIGURE 13. Pump with suppressor and regulator displacement PSD with supply pressure at 4 MPa. Root total CPS value is 0.6 nm.

Further improvements at the tooth pass frequency were realized with the addition of the high pressure oil regulator as shown in Figure 12 and Figure 13.

Finally, the spectra of Figure 8 and Figure 9 were used to create a frequency dependent pulsation sensitivity function shown in Figure 14. This response function is a result of the dynamic behavior of the bearing film and the structural compliance. The static pulsation sensitivity results are consistent with the low-frequency region of the function.

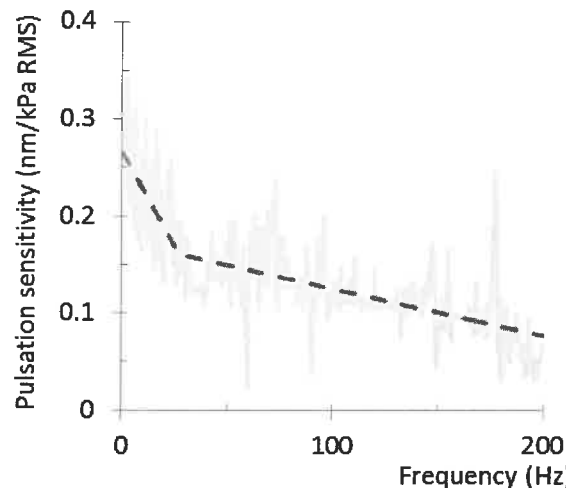


FIGURE 14. Frequency dependent pulsation sensitivity function shows the axial error due to pulsations from 1 Hz to 200 Hz. It is a combination of the dynamic behavior of the bearing compensation and the structural compliance of the rotor.

CONCLUSION

Hydrostatic spindles and slides exhibit errors when subjected to pressure ripple. Structural compliance is the dominant contributor to pulsation sensitivity at low frequencies. Design and testing of a high pressure oil supply that results in minimal spindle errors is described. Best results are obtained using a suppressor and regulator close to the pump with pressure pulsations of 1 kPa giving rise to axial displacements of 0.6 nm.

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LOCALIZED GROWTH AND FORCE DETECTION OF CARBON NANOTUBES ON MULTI-AXIS MEMS SENSOR

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ABSTRACT

This paper introduces the design and fabrication of a multi-axis force sensor for localized growth of carbon nanotubes (CNTs). A micro-heater is located at the center stage that joins three cantilever structures. This pairing generates a highly sensitive tool for the growth of CNTs. The force sensor resolution and range are 77.1 pN and 17.3 μ N respectively which allows the device to sense the reaction force of a CNT with a surface.

INTRODUCTION

One of the most popular methods for growing CNTs is chemical vapor deposition (CVD). While this method is common, high temperatures during the process limit what pre-existing materials can be on the substrate [1]. A solution to the integration of CNTs was first demonstrated by Englander who used Joule heating on a suspended silicon microbridge to isolate the growth CNTs and silicon nanowires [2]. Other researchers have used cantilever beams, micro heaters, and a range of resistive materials to act as heat sources [1]. All of these designs are limited by allocating valuable area on the wafer and additional processing steps to only grow CNTs in predefined locations. By moving the heat

source to the center stage of a multi-axis force sensor a precise, accurate, and repeatable fabrication process can be developed.

The CNT deposition system presented in this paper consists of a multi-axis MEMs force sensor with an integrated micro-heater, which provides the heat energy for the CNT grow. A mesoscale nanopositioning system is used to deposit the CNT on a surface. The force sensors are used to register the reaction force of the growing CNT when it makes contact with the surface and control the tension applied during positioning. This system is illustrated in Figure 1.

For CNT growth, a metallic catalyst is placed on pyramidal structure similar to an AFM tip located at the center of the stage joining three cantilever beams. The full sensor design is discussed in the next section. The sensor is then placed in a quartz tube, Figure 1.A. Power is supplied to the micro-heater to heat up the catalyst as hydrocarbons are introduced to the tube. Decomposing hydrocarbons initiate the growth of CNTs from the tip of the pyramid, Figure 1.B. Figure 1.C shows the CNTs growing towards and contacting the surface below, which is registered by the force sensor. Once the contact force is achieved, a nanopositioner travels at a constant

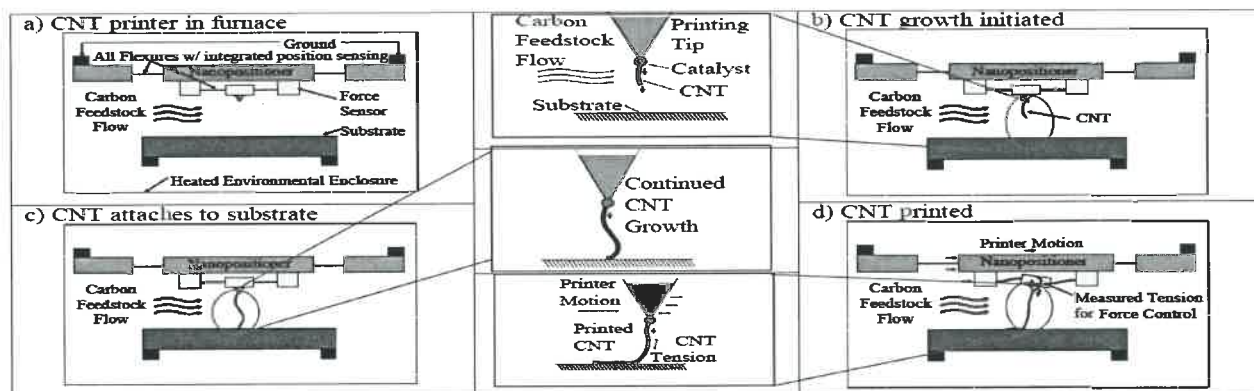


Figure 1. Direct printing of CNT to substrate

rate to maintain a tension in the CNT for precise positioning, Figure 1.D.

FORCE SENSOR

The multi-axis force sensor in Fig. 2 is composed of three symmetric cantilever beams connected by a center stage and fixed to wafer creating a fixed-guided system. Three beams allow for stage to travel in the vertical Z-axis and rotation about the X and Y axes to be measured by pairs of polysilicon n-type piezoresistor placed at the base of each beam. Integrated on the center stage is a pyramid and meander micro-heater. The pyramid creates a sub 100nm reference point for the growth origin of the CNTs. A tungsten micro-heater acts as the heat source for the CVD process. In addition to separating the growth process from the desired substrate, the localized heating allows for a room temperature environment inside the furnace.

Design

In this application, the force resolution and dynamic range of the sensors are the primary design parameters. The upper limit of the force resolution was set by the 0.16 nN growth force of a CNT. The growth force was measured by placing tungsten weights on top of catalyst substrate prior to the CVD process. The CNT bundle lifted the weights during the growth process resulting in a growth force [3]. Dynamic range, the ratio of force range to resolution, of the sensor is optimized by applying a system noise level model developed for MEMs sensors systems [4].

The model defines the three dominate sources of noise in its system: Johnson noise, flicker noise, and instrumentation amplifier noise. Johnson noise is a response to the thermal agitation of electrons in the resistor. Flicker noise is produced by the fluctuations in charge carriers. The dynamic range, DR, equation combines these noise sources from the piezoresistor with the instrumentation noise to be written as:

$$DR = \frac{\sigma_y N_E G_F G_{STC} V_S G_{SG}}{\eta_{EM} \sqrt{4k_B T R B + (V_S^2 / 16) \frac{\alpha}{C_c \Omega} \ln\left(\frac{f_{max}}{f_{min}}\right) + S_{Vai} B}} \quad (1)$$

In Eq. 1, G_F , G_{SG} , R , and Ω can be written in terms of the cantilever beam and piezoresistor dimensions, supply voltage, and carrier concentration [4]. Constraints for the optimization

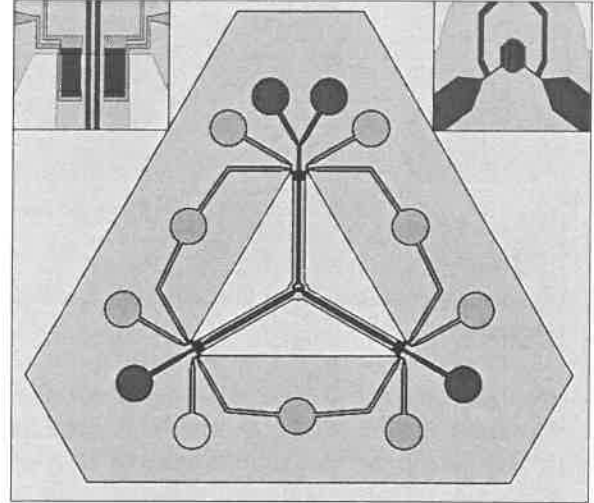


FIGURE 2. Top view of multi-axis force sensor with a zoomed in view of the piezoresistors in the top left corner and the micro-heater in the top right corner

are defined by the system, available manufacturing process, geometry, and force resolution. By selecting polysilicon as the piezoresistive material, the initial gauge factor, G_F , carrier concentration, and resistivity could be defined. The G_F value was maximized at 20 for n-type doping. Doping concentrations and resistivity were determined to be 3×10^{19} atoms/cc and 0.01 Ω -cm [5].

The force resolution, F_{res} , equation is the maximum force divided by the DR. In Eq. 2, ϵ_F is the force gain for the flexure [4].

$$F_{res} = \left(\frac{\sigma_y}{\epsilon_F \cdot SF} \right) / DR \quad (2)$$

The F_{res} value is adjusted to match or exceed the growth force of a CNT. As the force resolution nears the pN range, thermomechanical noise must be introduced as the lower limit for F_{res} to maintain the accuracy of the sensor [6].

$$F_{thermomechanical} = \sqrt{\frac{2k_B T B}{\pi Q \omega_n}} \quad (3)$$

Fabrication Process

The force sensor is fabricated using the available processes from the Microelectronics Research Center and Center for Nano – and Molecular Science at The University of Texas at Austin. A 100 mm double silicon-on-insulator (SOI) wafer is used as the substrate to fabricate the sensor. The SOI wafer is composed of two 2 μ m device layers and one 450 μ m handler layer of silicon

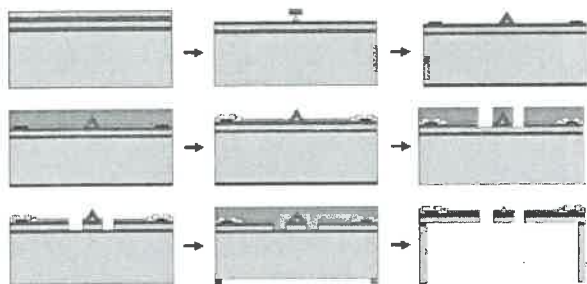


Figure 3: Process flow for the force sensor fabrication

separated by two 0.5 μm silicon dioxide layers illustrated in first image in Figure 3. First, the pyramid is formed via wet etching with KOH+IPA of 2 μm circular features. Second, 198 nm amorphous silicon layer is deposited with low pressure CVD (LPCVD). Diffusing doping with POCl_3 at 900°C is used to dope with amorphous layer of amorphous silicon is deposited with LPCVD and doped in a POCl_3 diffusion furnace. During this process, the amorphous layer transitions into polycrystalline silicon with a phosphorous concentration of $3 \times 10^{19} \text{ atoms/cc}$. Piezoresistors are defined with a dry etch. The tungsten micro-heater is patterned next using a liftoff process. A liftoff process deposits the material, tungsten, over the patterned photoresist. Excess material is lifted off when the photoresist layer removed with acetone. Next, chromium layer is deposited and patterned for the bond pads and wire traces to the piezoresistors.

With all of the sensor's featured created, the cantilever beams can be defined and etched from the front side with reactive ion etching (RIE). Deep RIE etching removes the 450 μm bottom layer suspending the cantilevers completely. To complete the release, the final oxide layer is removed.

MICRO-HEATER

The micro-heater is a 200 nm tungsten meander design. Wire traces are connected before and after the heater to create a four-wire sensor shown in Figure 4. The larger traces supply current to the heater. A change in voltage is measured with the two smaller traces. The known temperature coefficient of resistance of tungsten can be used to convert the change in voltage across the heater into a measurement of the

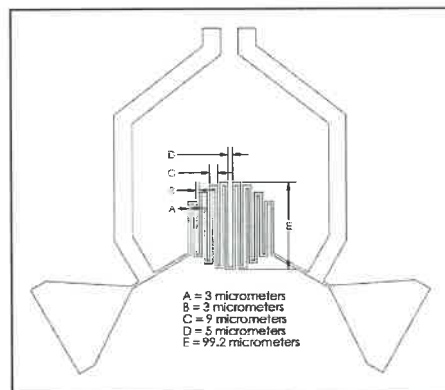


Figure 4: Schematic of meander micro-heater with 4-wire design.

change in temperature of the heater. Once calibrated, the current supply can be adjusted to accurately control the CVD temperature.

Design

The design of the micro-heater is broken into two parts: material selection and dimensions. The temperatures required for growth of multiwalled (MWCNT) and singled walled (SWCNT) CNTs provide the material properties needed. The temperature range required to use CVD for CNTs is 600°C to 1200°C. MWCNTs are typically grown in the 600°C to 900°C range, while SWCNT require temperature from 900°C to 1200°C [7]. To stably operate the heater at these temperature, a material with a high melting point and non-degraded electrical resistance is required. Out of the silicon and metallic based options, tungsten is most suitable choice. It has a melting point of 3140°C and maintains a linear temperature coefficient of resistance. Also there are standard deposition and etching processes available [8].

As for the dimensions, the film thickness must be enough to promote adhesion, but thin enough to require a small power supply. The heater line width, 3 μm , is driven by a resistance ratio, cantilever beam width, and heater area. A ratio of 20:1 heater to wire trace resistance is used to reduce the impact of the wire traces on the power supplied to the heater. With the piezoresistors and chromium wire traces at the base of the beam, the tungsten wire trace width is limited to 110 μm . To minimize thermal effects of the heater on the uniformity of the cantilever deformations, the heater was limited to 100 μm circular field. These constrains lead to the 3 μm modified

meander design. The modified meander design generated a heater length of 1.398 mm resulting in an electrical resistance of 805 Ω .

Simulation

A static heat transfer simulation have been conducted to determine the heat distribution over the device. COMSOL™ 4.3b is used to study two models of the sensor. Symmetry is used to reduce size of the models and computational time. For the power required for the heater, half of the SOI wafer and micro-heater are modeled. A third of the SOI wafer, the micro-heater, and two of the piezoresistors are modeled to determine temperature gradient across the resistors with a power supply of 0.1 W. The SiO₂ layers were excluded due to its low thermal conductivity.

Under the heat transfer in solids study, a convective heat transfer coefficient of 5 W/m²K is applied to all surfaces except the base and the sections cut for symmetry. The temperature of the base is set to room temperature 300 K. A heat source is applied either to the heater or the piezoresistors. For the heater, the magnitude of the heat source is adjusted in terms of power until the center of the heater reached 1300 K. Figure 5 shows the heat distribution from the center stage to the surrounding silicon structure. When a power of 0.0425 W is applied to the heater the center reached 1292.9 K and is dissipated to 300 K by the end of the cantilever beam. That power supply, 0.085W for a whole heater, can be used with the wire trace and heater resistances to calculate the voltage supply, which is 8.27 V.

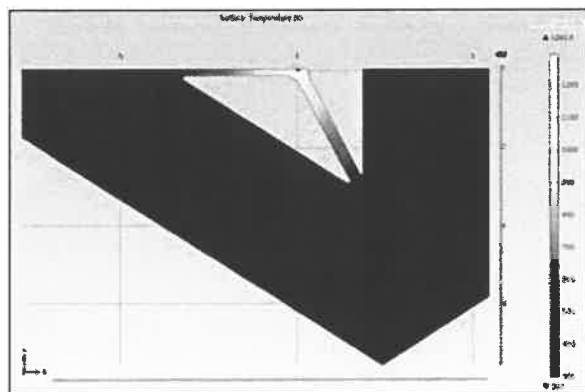


Figure 5: One third sensor with 0.08 W power supply to micro-heater.

The next study supplies 0.1 W of heat power to each piezoresistor to simulate operating conditions. At the center stage, the heater temperature was defined to 1300 K. Figure 6 shows the temperature range from 300 K to 343.5 K along the resistor.

RESULTS

The noise optimization method produced a force sensor design where the DR and F_{res} are 107.02 dB and 77.1 pN respectively. The derived dimensions for the cantilever beams result in a combined stiffness of 0.042 N/m. Resistance of the piezoresistor is 773.95 Ω . The supply voltage is 8.797 V, which results in an operating power of 0.1 W per piezoresistor. Dimensions for the cantilever beam and piezoresistors are shown in Table 1.

Table 1. Flexure and piezoresistors dimensions.

(μ m)	Length	Width	Height
Flexure	3000	324.2	2
Piezoresistor	130.2	85.1	0.198

The simulations show that the handler layer surrounding the suspended cantilever beams dissipates the heat produced by the micro-heater. This should limit the amount of heat applied to the printing surfaces. With the operational power supply, the piezoresistors have a 43.5 K temperature gradient. Similar to the temperature monitoring of the micro-heater, the known temperature coefficient of resistance can be applied to maintain adequate supply voltage.

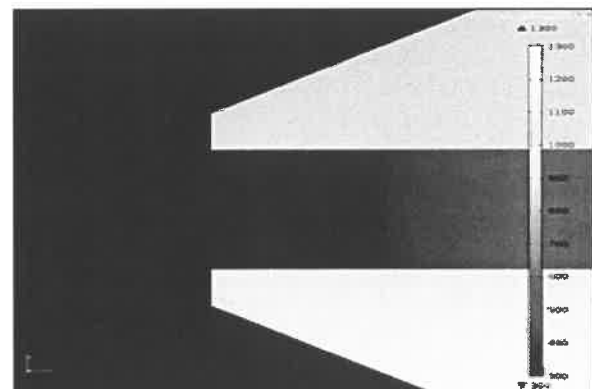


Figure 6: One third sensor with 0.1W power supply to piezoresistors.

CONCLUSION

The force sensor design developed in this paper has a force resolution which is one third of the anticipated growth force. This design is the first step towards the direct printing of CNTs with high precision and accuracy onto substrates with temperature sensitive materials. This introduces a new process for integrating sub 20 nm features with desirable electrical, mechanical, thermal, and optical properties. The metallic form of CNTs can be used as robust wire traces on flexible devices with an elastic modulus in the 1 TPa range [7]. CNTs can also be used as transducers in nanoscale devices such as field effect transducers, piezoresistive sensors, and infrared sensors [9]. Finally, by growing and positioning the CNTs on to the substrate with the force sensor, the number of lithography and etching processes are reduced for a given device.

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SYNTHESIS OF THERMAL ERRORS IN GIANT FLOOR TYPE BORING MACHINES

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ABSTRACT

Based on black-box type identification method, dynamic thermal error models of shape and joint elements in giant floor type boring machines are obtained as discrete models. To derive error transfer functions (TFs) of the error elements, input and output data of the dynamic subsystems have been collected through the finite element (FE) thermal analysis. Synthesis of thermal error in the giant floor type boring machine is conducted by using the TFs and the previously developed kinematic error chain [1-2].

INTRODUCTION

Giant machine tools are applied for mega size machining in the ship building, aerospace and variety of plant industries. As the machining parts are huge and too much machining time is required, volumetric errors due to thermal deformation of the shape and joint elements degrade machining accuracy.

Kim and Chung [3], studied modeling of volumetric errors due to geometric and thermal deformation errors in small-size machine tools. To predict volumetric errors in the design process of giant floor type boring machines, kinematic error modeling was conducted by shape and joint transformation matrices [1]. Thermal deformation TFs of the shape and joint elements according to internal and external heat sources were formulated for 3D time varying error analysis in the giant floor type boring machine [2]. However, the error elements were obtained from heuristic methods. To obtain accurate thermal error models for precise and rapid design of giant machine tools, black-box type dynamic modeling of the error TFs through data acquisition of heat inputs and deformation output data is required.

In this paper, to describe the identification procedure of the dynamic error models, the column unit shown in Fig. 1 is selected an example. Similar procedures are applied for

other units to determine deformation TFs. To collect input and output data, a lot of FE thermal analysis with proper boundary conditions is performed in this paper. Ambient temperatures according to height are used for external heat sources. Heat generated from the ballscrew interface, support bearings, motors, gears, spindle bearings and sliding units, and cooling units are internal heat sources. They depend on operating conditions. Convections on the head unit and the base plate are formulated as boundary conditions. TFs of the error elements are identified by discrete models based on black-box approach [4].

VOLUMETRIC THERMAL ERROR MODEL

Fig. 1 shows schematic diagram of the giant floor type boring machine. Column base, head frame, rotary base, and so on are shape elements. Moving and rotary elements such as rotary table, floor bed, column, and ram are joint elements. Shape transformation matrices consist of time varying error terms. Joint transformation matrices depend upon time varying and spatial thermal error terms as well.

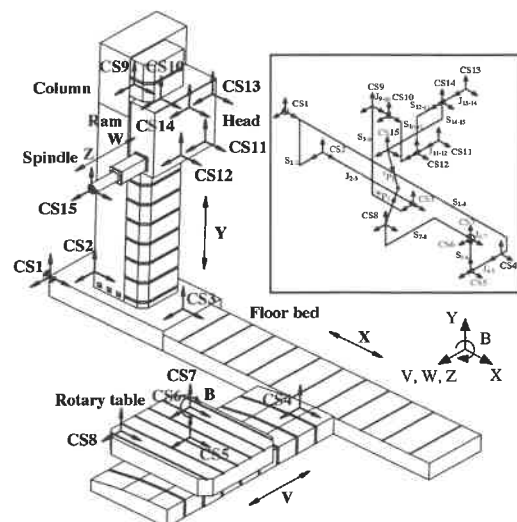


FIGURE 1. Schematic diagram of a giant floor type boring machine.

Eq. (1) shows shape and joint transformation matrices of the column with geometric and thermal deformation errors [1-2]. ε_{x3}^S , ε_{y3}^S and ε_{z3}^S are angular errors of shape transformation matrices. ε_{x3}^J , ε_{y3}^J and ε_{z3}^J are angular errors of joint transformation matrices. Δa_3 , Δb_3 and Δc_3 are translational errors due to thermal deformation of shape elements. S_{xy} , δ_x , δ_y and δ_z are squareness, positioning and straightness errors. S_{xy}' , δ_y' and δ_z' are squareness, positioning and straightness errors due to thermal deformation. P_y is thermal expansions in the feed axis.

$$[S_{3 \rightarrow 9}]_{actual} = \begin{bmatrix} 1 & -\varepsilon_{z3}^S(T) & \varepsilon_{y3}^S(T) & a_3 + \Delta a_3(T) \\ \varepsilon_{z3}^J(T) & 1 & -\varepsilon_{x3}^S(T) & b_3 + \Delta b_3(T) \\ -\varepsilon_{y3}^J(T) & \varepsilon_{x3}^J(T) & 1 & c_3 + \Delta c_3(T) \\ 0 & 0 & 0 & 1 \end{bmatrix} \quad (1)$$

$$[J_{9 \rightarrow 10}]_{actual} = \begin{bmatrix} 1 & -\varepsilon_{z4}^J(T) & \varepsilon_{y4}^J(T) & \delta_z + S_{xy} \cdot y + S_{yz}(T) \cdot y \\ \varepsilon_{z4}^J(T) & 1 & -\varepsilon_{x4}^J(T) & y + \delta_y + \delta_y'(T) + P_y(T) \cdot y \\ -\varepsilon_{y4}^J(T) & \varepsilon_{x4}^J(T) & 1 & \delta_z + \delta_z'(T) \\ 0 & 0 & 0 & 1 \end{bmatrix}$$

Considering dominant thermal error terms only, volumetric errors at the tool center point (TCP) due to the column in X, Y and Z directions are given by [2]

$$\begin{aligned} dX &= \Delta a_3(T) + S_{xy}(T) \cdot y \\ dY &= \Delta b_3(T) + P_y(T) \cdot y \\ dZ &= \Delta c_3(T) + \delta_z'(T) \end{aligned} \quad (4)$$

NUMERICAL SIMULATION

In previous papers [1-2], error terms in the shape and joint error matrices were formulated through heuristic approaches. As the machine tool is huge, heuristic modeling by experiment is difficult and expensive. In this paper, thermal error elements are modeled through FE analysis. Fig. 2 shows FE model of the column. It is divided into 20 parts. Following assumptions are applied for the thermal analysis: i) Corner of the bottom surface is fixed. ii) Bottom surface deforms freely in X and Z axes.

Boundary conditions of the joint element are derived from frictional heat and convection coefficient of the ballscrew unit [5]. Generated heats due to the moving nut, fixed bearings and

the spindle head unit are applied for the analysis as internal heat sources. Ambient temperatures shown in Fig. 3 (a) are external heat sources. Fig. 3 (b) shows internal heat source as a step input. Fig. 4 shows heat input to verify the identified thermal error models.

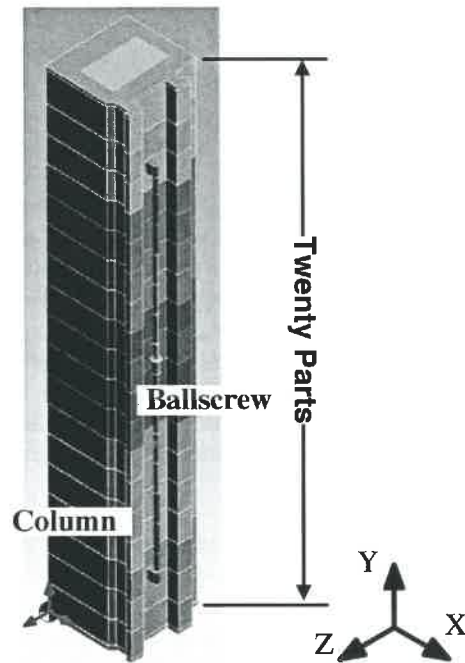
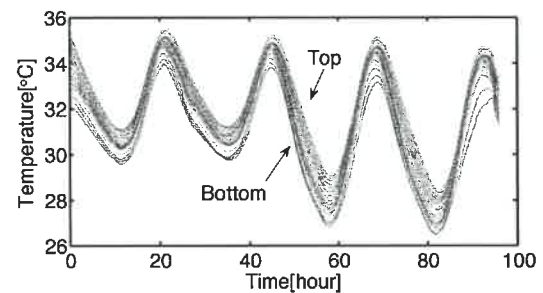
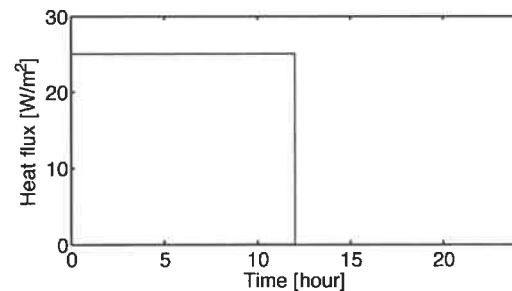


FIGURE 2. Finite element model of the column.

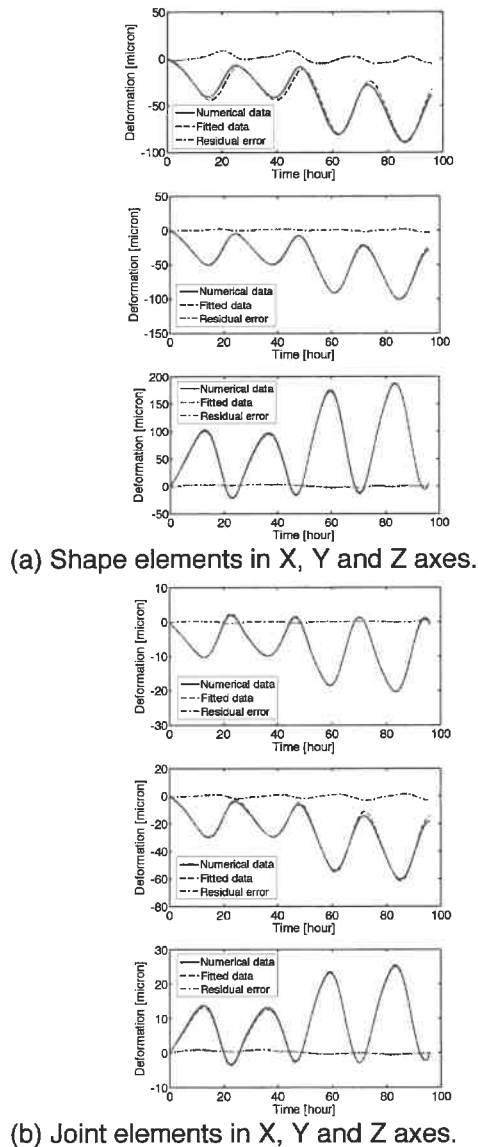
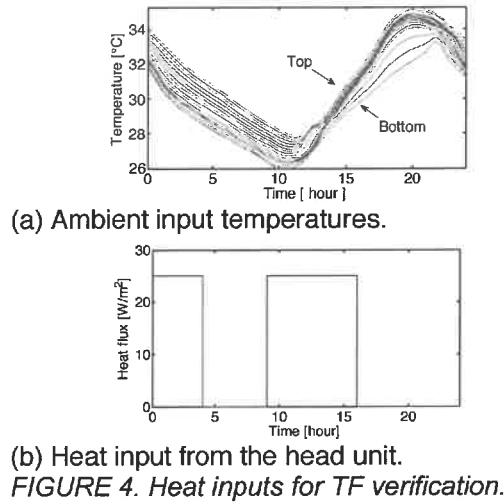


(a) Ambient input temperatures for 96 hours.



(b) Heat input from head unit for 24 hours.

FIGURE 3. Heat inputs for TF identification.



IDENTIFICATION OF THERMAL TF MODELS

To identify thermal deformation TFs in discrete domain, TCP thermal errors are separated into environmental and operational terms as follows:

$$\begin{aligned} dX &= {}^E\Delta a_3(T) + {}^O\Delta a_3(T) + {}^E S_{xy}(T) \cdot y + {}^O S_{xy}(T) \cdot y \\ dY &= {}^E\Delta b_3(T) + {}^O\Delta b_3(T) + {}^E P_y(T) \cdot y + {}^O P_y(T) \cdot y \\ dZ &= {}^E\Delta c_3(T) + {}^O\Delta c_3(T) + {}^E\delta_z(T) + {}^O\delta_z(T) \end{aligned} \quad (5)$$

Fig. 5 shows FE simulations and estimated environmental TF models. Residual errors are less than 10µm. Environmental shape error TF for ambient temperature input $\theta(T)$ in X-axis is given by

$${}^E\Delta a_3(T) = \frac{0.2471z^{-1}}{1 - 0.9926z^{-1} + 0.007289z^{-2}} \theta(T) \quad (6)$$

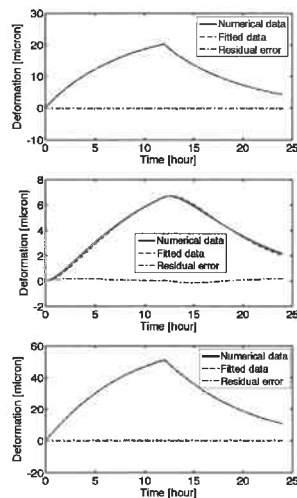
Converting this to continuous time domain, unit step shape and joint error responses are obtained as follows:

$$\begin{aligned} {}^E\Delta a_3(T) &= \sum_{i=1}^2 {}^E D_{ai}^s \left(1 - e^{(-t/\tau_{ai}^s)} \right) = 16.83 \left(1 - e^{(-t/134)} \right) - 0.0019 \left(1 - e^{(-t/0.4)} \right) \\ {}^E\Delta b_3(T) &= \sum_{j=1}^1 {}^E D_{bj}^s \left(1 - e^{(-t/\tau_{bj}^s)} \right) = 18.04 \left(1 - e^{(-t/24.45)} \right) \\ {}^E\Delta c_3(T) &= \sum_{k=1}^3 {}^E D_{ck}^s \left(1 - e^{(-t/\tau_{ck}^s)} \right) = 114.8 \left(1 - e^{(-t/6.67)} \right) + 24.45 \left(1 - e^{(-t/339.6)} \right) \\ &\quad - 163.375 \left(1 - e^{(-t/1.3)} \right) \\ {}^E S_{xy}(T) &= \sum_{i=1}^2 {}^E D_{xi}^j \left(1 - e^{(-t/\tau_{xi}^j)} \right) = 5.94 \left(1 - e^{(-t/2.22)} \right) - 3.09 \left(1 - e^{(-t/228)} \right) \\ {}^E P_y(T) &= \sum_{j=1}^1 {}^E D_{yj}^j \left(1 - e^{(-t/\tau_{yj}^j)} \right) = 10.78 \left(1 - e^{(-t/45.22)} \right) \\ {}^E\delta_z(T) &= \sum_{k=1}^3 {}^E D_{zk}^j \left(1 - e^{(-t/\tau_{zk}^j)} \right) = 112.95 \left(1 - e^{(-t/53.49)} \right) + 62.7 \left(1 - e^{(-t/61.76)} \right) \\ &\quad - 178.45 \left(1 - e^{(-t/52.91)} \right) \end{aligned} \quad (7)$$

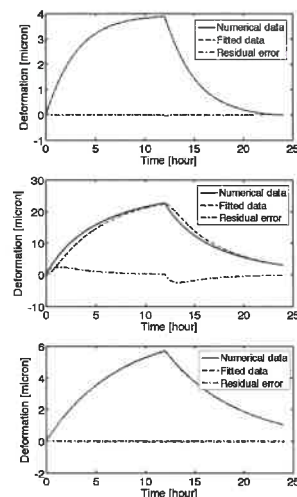
Error TF consists of several first order systems. Time constants and gains denote thermal modes and error magnitudes, respectively.

Fig. 6 shows FE analysis and TF model results according to operating conditions. Maximum residual error is less than 5µm. Equation (8) shows shape and joint error TF terms according to operating conditions in the continuous time domain. They consists of several first order systems as well. Gains mean error magnitude of each thermal deformation mode. Time constants denote eigenvalues of the thermal modes.

$$\begin{aligned}
{}^o\Delta a_3(T) &= \sum_{i=1}^2 {}^oD_{ai}^s \left(1 - e^{(-t/{}^o\tau_{ai}^s)}\right) = 8.68 \left(1 - e^{(-t/74.07)}\right) - 0.0013 \left(1 - e^{(-t/0.37)}\right) \\
{}^o\Delta b_3(T) &= \sum_{j=1}^1 {}^oD_{bj}^s \left(1 - e^{(-t/{}^o\tau_{bj}^s)}\right) = 3.26 \left(1 - e^{(-t/215.6)}\right) \\
{}^o\Delta c_3(T) &= \sum_{k=1}^3 {}^oD_{ck}^s \left(1 - e^{(-t/{}^o\tau_{ck}^s)}\right) = 13.15 \left(1 - e^{(-t/1.73)}\right) + 4.59 \left(1 - e^{(-t/95.09)}\right) \\
&\quad - 0.00079 \left(1 - e^{(-t/95.09)}\right) \\
{}^o\delta_{xy}(T) &= \sum_{i=1}^2 {}^oD_{xi}^t \left(1 - e^{(-t/{}^o\tau_{xi}^t)}\right) = 2.37 \left(1 - e^{(-t/1.49)}\right) - 0.95 \left(1 - e^{(-t/306)}\right) \\
{}^oP_y(T) &= \sum_{j=1}^1 {}^oD_{yj}^t \left(1 - e^{(-t/{}^o\tau_{yj}^t)}\right) = 9.51 \left(1 - e^{(-t/36.7)}\right) \\
{}^o\delta_z(T) &= \sum_{k=1}^3 {}^oD_{zk}^t \left(1 - e^{(-t/{}^o\tau_{zk}^t)}\right) = 1.85 \left(1 - e^{(-t/2.34)}\right) + 0.079 \left(1 - e^{(-t/82.46)}\right) \\
&\quad - 0.00011 \left(1 - e^{(-t/82.46)}\right)
\end{aligned} \tag{8}$$



(a) Shape elements in X, Y and Z axes.



(b) Joint elements in X, Y and Z axes.

FIGURE 6. Identified operational TF models.

VERIFICATION OF THERMAL ERROR TFs

Identified environmental and operational TF models are combined to estimate total thermal errors as shown in Fig. 7. It confirms that

developed thermal deformation TF models estimate thermal error well.

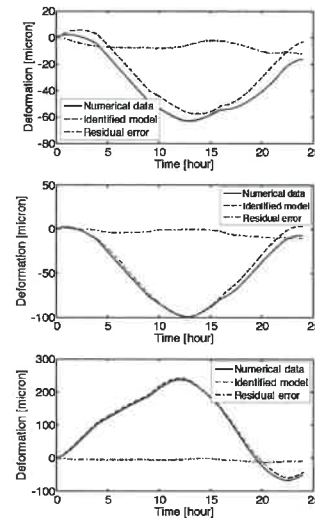


FIGURE 7. Thermal errors in X, Y and Z axes.

CONCLUSIONS

Dynamic modeling of thermal error in the giant machine tool has been conducted through kinematic error modeling and black-box type system identification method. Heat inputs and output thermal error data are collected from FE analysis. Discrete type error TFs of shape and joint elements are identified through output error models. It is confirmed that thermal error of the giant machine tool is estimated well through the dynamic thermal error modeling.

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PROCESS FOR DEVELOPING STAKEHOLDER-DRIVEN REQUIREMENTS AND CONCEPT OF OPERATIONS

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INTRODUCTION

This paper describes a process for developing the requirements and concept of operations for an engineered system that are traceable to the mission goals and stakeholder needs. The determinism provided by this up-front investment during the project life cycle sets the stage for developing a solution that is responsive and traceable to the goals and needs. An overview is provided on systems architecting and systems engineering and how they fit into the larger role of project management, and a view of a project life cycle from a combined systems architecting and engineering perspective is presented.

MOTIVATION

When developing a system to satisfy a mission need or opportunity, establishing correct and complete requirements and a concept of operations (ConOps) enables informed design decisions and improves the likelihood of long-term project success. A sufficiently rigorous approach to connecting mission goals and stakeholder expectations to the design of a new system can help a project team avoid the trap of pursuing a solution that only they see as being important and ultimately failing due to the loss of sponsorship or not meeting the needs of a key end-user, support infrastructure, or interface. Referring to Figure 1, the solution development phase of a project is connected to the mission

goals and stakeholders by the system requirements and concept of operations. Any discussion or examination of a proposed solution that is not framed in the context of the requirements and concept of operations runs the risk of being disconnected from the reasons and benefactors of pursuing the project.

SYSTEMS ARCHITECTING & ENGINEERING

Systems architecting and engineering are sometimes referred to collectively as “systems engineering”. They overlap with each other but encompass two distinct areas of focus that are vital to project success [1, 2, and 3]. Systems architecting supports the stakeholders in identifying the mission goals and requirements, and exploring solution paths. It is not uncommon for an end-user or customer to be unclear about what is needed, in which case they need an advocate to help them determine those needs before momentum builds towards a particular solution. Systems engineering creates and manages the technical plan for developing and delivering a best solution, and supporting its entire life cycle. A combined systems architecting and systems engineering view of a project life cycle is shown in Figure 2. The majority of the project life cycle cost is committed when turning a conceptual design into a final design, so the up-front investment in systems architecting and systems engineering activities is important.¹ Systems architecting and engineering overlap with project control, which ensures project execution according to governing processes, and all three lie within the larger role of project management, which seeks a self-consistent balance of scope, schedule, budget, performance, and safety. The remainder of this paper focuses on working through the first two major phases in Figure 2, to establish



FIGURE 1. Phases in a systems architecting and engineering –centric view of a project life cycle. Mission goals and stakeholders form the foundation upon which the two pillars of requirements and concept of operations stand, providing a solid base for solution development.

¹ A 1993 study of U.S. Department of Defense projects shows that 80% of a project's life cycle cost is committed by the initial 20%, and that the cost to extract defects from a system increases geometrically during the project life cycle [4].

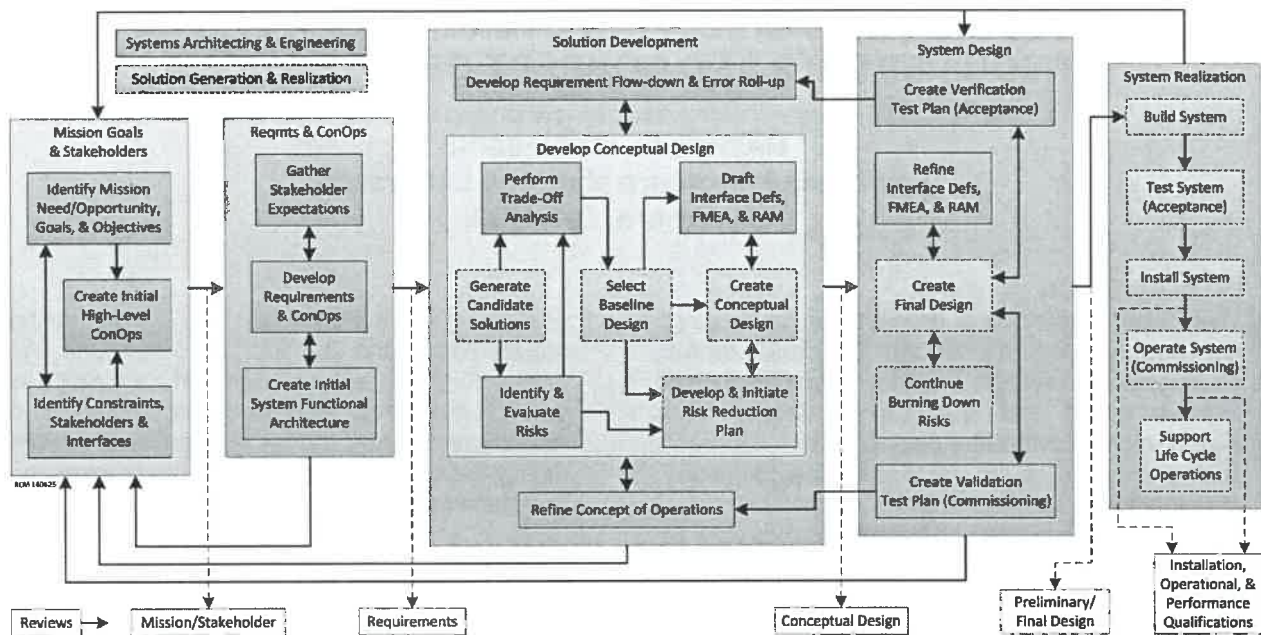


FIGURE 2. A systems architecting and engineering –centric view of a project life cycle.

requirements and a co-evolved concept of operations that are connected to the mission goals and stakeholders.

DEVELOPING REQUIREMENTS AND A CONCEPT OF OPERATIONS

The process for going from an initial mission definition to a requirements review, which includes the concept of operations, is broken down into the five phases shown in Figure 3 and is shown in schedule format in Figure 4. A graded-approach to developing requirements and a concept of operations is achieved by adjusting the depth of engagement within each phase, and the number of iterations within and between phases, to an appropriate level relative to the mission importance and project size.

Need or Opportunity Definition

At the outset of a project it is critical to establish a clear vision of the need or opportunity, goals, metrics for success, and constraints. Identifying a single mission need or opportunity helps avoid conflicting goals. Mission goals are qualitative high-level expectations for the system, and one or more objectives flow-down from each goal as quantitative targets that define how success will be determined [5]. Explicitly stating the project constraints, like schedule, budget, management or regulatory, and technology, provides clarity between the project team and stakeholders. Engaging key stakeholders during this phase

provides the domain knowledge needed for accuracy and completeness, and establishes a shared understanding that can be referred to during project execution when the questions arise, “Why are we doing this, Why is it taking so long, and Why is this so hard?” A mission review assesses project-start feasibility by looking for consistency between the goals and constraints, enabling early schedule and budget adjustments and informing needed trade-offs between goals (see Table 2).

Operational Context

Knowing how a system fits into the rest of the world and how it will be used informs what it needs to be. A system context diagram, like the example shown in Figure 5, illustrates the boundary of the system, stakeholders, external interfaces, and key influences. Clearly defining the boundary between the system and its environment helps establish the scope of the project. Placing the stakeholders around the periphery of the diagram is analogous to gathering everyone around a table to discuss a shared big-picture view of the project. The system context diagram provides a basis for developing the operational scenarios for the system, from which the initial high-level concept of operations is derived. Qualitative functional, operational, and safety requirements are discovered by stepping through the concept of operations and asking, “What is needed to

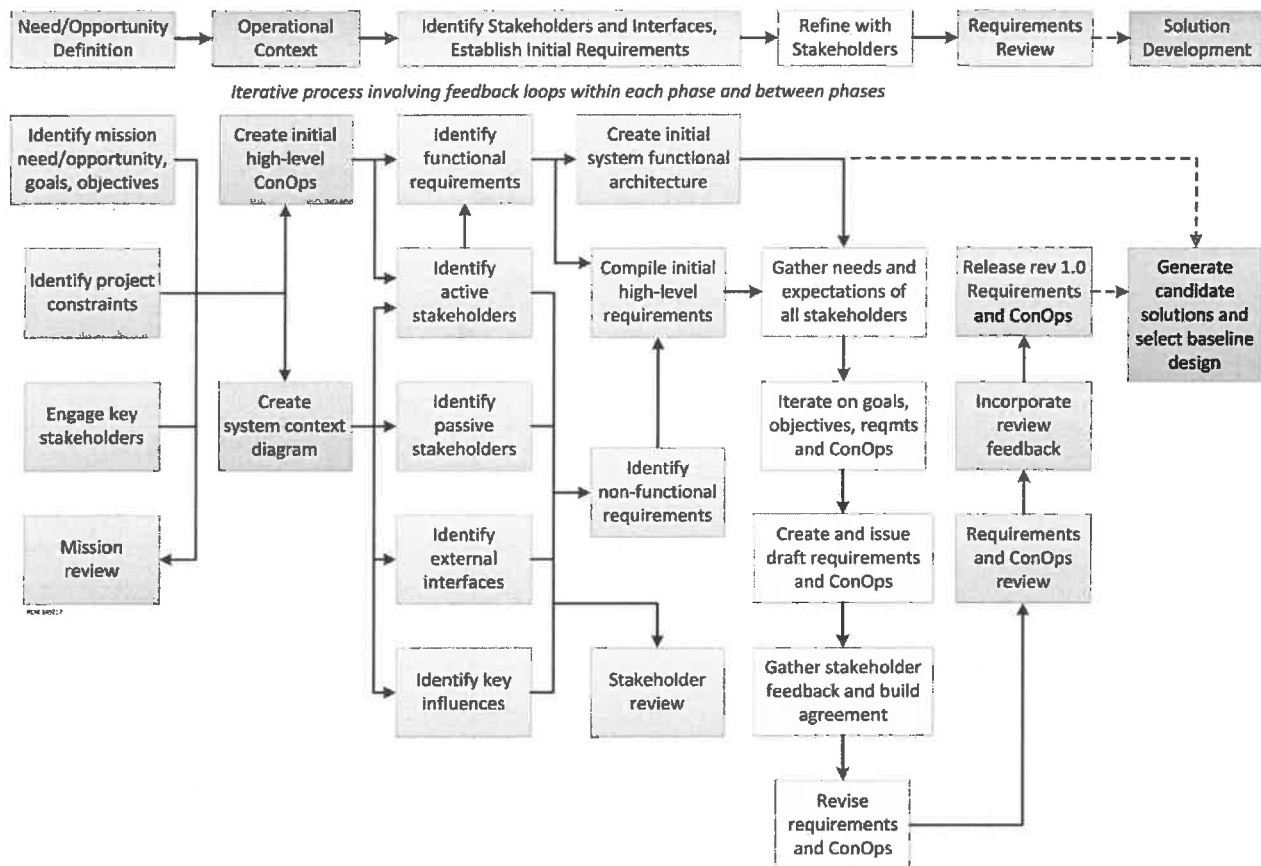


FIGURE 3. Process for developing stakeholder-driven requirements and concept of operations. Iterations within each phase (vertical swim lanes) and between phases are not shown. The dotted lines connect the output of this process to the third major phase (Solution Development) in Figure 2.

perform this step?" Typically, each of the resulting requirements flows down to one or more quantitative performance requirement.

Stakeholders, External Interfaces, and Initial Requirements

As the system context diagram and operational scenarios are developed, the list of stakeholders and external interfaces evolves through a process of discovery and refinement. Engaging the stakeholders^b that will interact with and be responsible for the deployed system informs the development of the concept of operations, its related requirements, and the validation plan. Additional high-level requirements are identified during discussions with key stakeholders, and drawn from experience with a similar mission or

^b "Active" stakeholders are responsible for and interact with the deployed system. "Passive" stakeholders influence the development and success of the system but do not directly interact with it [5].

system context. Grouping all the requirements by type as shown in Table 1 focuses stakeholder discussions, reveals omissions, and enables reuse between projects. To ensure adequacy during this phase, it is helpful to maintain a (growing) list of stakeholder domains and external interfaces, and to identify the people who will represent and speak for them during discussions. By the end of this phase a clear vision can be articulated regarding the mission need or opportunity, goals, objectives, constraints, system context, and initial high-level requirements and concept of operations. Establishing a broad view of the system and who will expect what from it helps prioritize and focus the more extensive stakeholder discussions during the next phase. A stakeholder review provides an early project-readiness assessment by looking for a big-picture view and engagement of the right set of people needed to develop correct and complete requirements and a concept of operations (see Table 2).

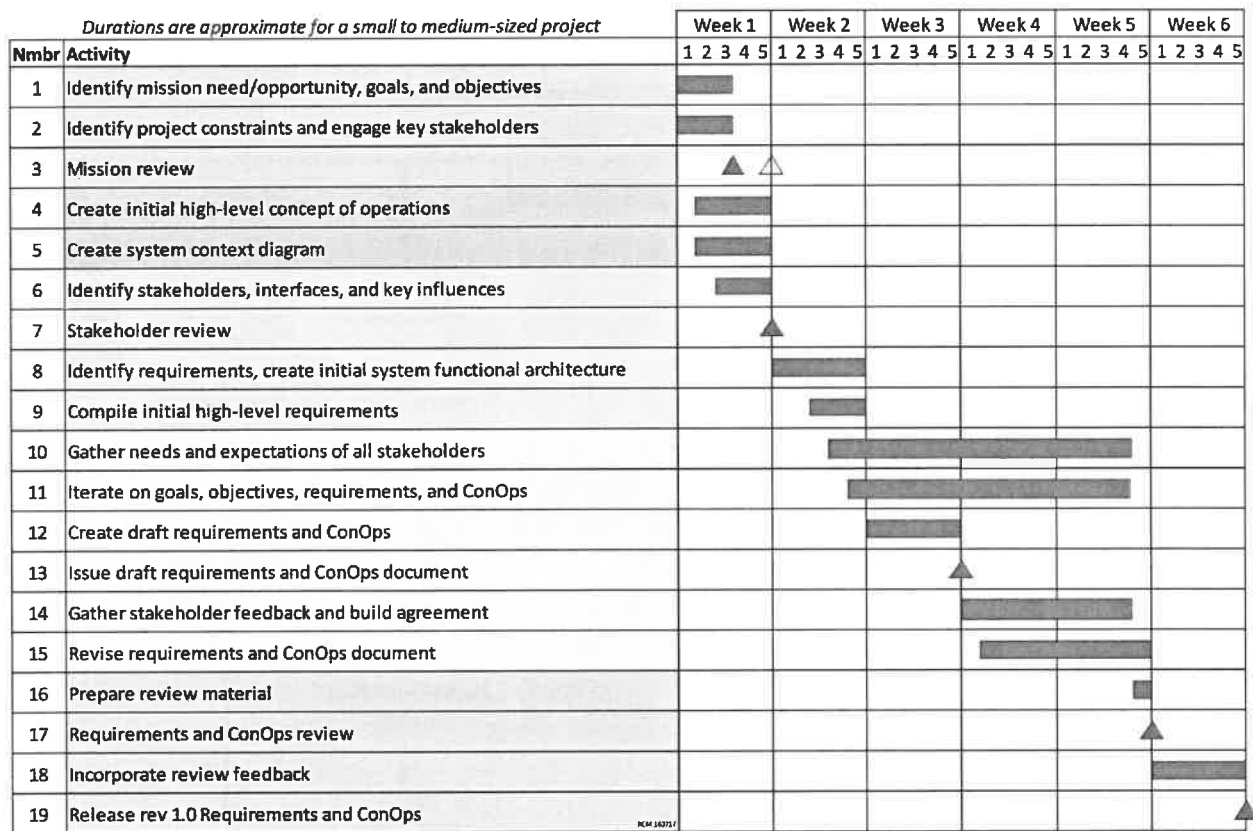


Figure 4. Schedule version of the process for developing stakeholder-driven requirements and concept of operations. Durations are approximate for a small to medium-sized project. The mission and stakeholder reviews can be combined (as indicated by the unfilled triangle).

Refining with Stakeholders

A systems architect/engineer's talents and skills for networking, listening, and being organized are fully called upon when gathering stakeholder needs, expectations, and feedback; building agreements; and sorting through, prioritizing,

and translating needs and expectations into refined versions of the requirements and concept of operations. Mapping stakeholders to the high-level requirements and concept of operations developed during the previous phase identifies whose voice needs to be heard when

0.0	Mission Need or Opportunity	Problem the system is supposed to solve or opportunity it targets
1.0	Mission Goals and Objectives	High-level expectations for the system (qualitative goals), and flow-down to how success will be determined (measurable objectives)
2.0	Functional and Performance Requirements	Functions the system needs to provide (qualitative), and flow-down to how well it needs to perform them (quantitative)
3.0	Interface Requirements	Interactions with and expectations on systems and activities outside the system boundary
4.0	Operational Requirements	Operational environment, human factors, RAM, FMEA, and recovery
5.0	Safety Requirements	Personnel health and safety, environmental protection, and equipment protection
6.0	Standards, Codes, Regulations, and Procedures Requirements	Established and approved practices, materials, processes, and protocols

Table 1. Suggested categories for organizing requirements. Although mission need/opportunity, goals, and objectives are not strictly requirements, including them in the requirements document provides traceability when judging the relative importance of a requirement to the mission.

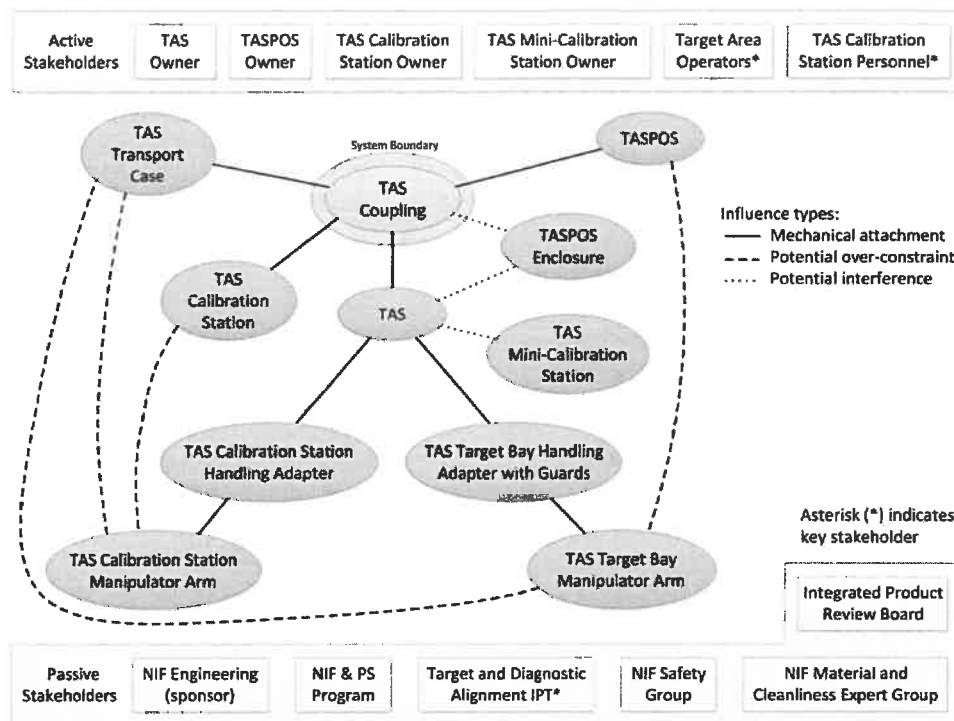


FIGURE 5. A system context diagram for the Target Alignment System (TAS) coupling for the National Ignition Facility (NIF). This simple example includes key influences between systems. The coupling provides a stable and repeatable attachment of the TAS to three other systems (one at a time).

adding to and refining them, and enables focused, efficient, and effective small-group discussions³ during stakeholder interviews. An important job for a system architect/engineer is ensuring that the voice of every stakeholder representative is heard, captured, and considered. If an important voice is not heard until later during the project life cycle, it could be in the form of a significant shortcoming of the deployed system that would have been less costly and disruptive to deal with sooner. Success during this phase requires stakeholder involvement and feedback during interviews and reviews of draft documents. Management's support of the importance of stakeholder engagement – making it a priority – improves the likelihood of a successful project. Striving for early feedback and rapid iterations helps make efficient use of everyone's time, and can be vectored towards issuing an early draft of the requirements and concept of operations to support the solution development phase.

³ Compared to gathering everyone in a room and "battling it out" until the requirements are declared done, which risks only the dominant voices being heard and completion by attrition.

Stakeholder feedback on draft documents identifies areas requiring refinement, additional discussion, and agreement building. Following an organized process and documenting as you go, especially keeping track of who provided input on what, identifies missing discussions, provides traceability, and helps meet an aggressive schedule.

Requirements Review

Before a project commits significant resources to solution development, all five phases preceding it in Figure 3 should be considered to an appropriate degree. There is an improved likelihood of success if this is done explicitly and with traceability. A suggested review committee charter for assessing the thoroughness and appropriate graded-approach for that effort is shown in Table 2. High-level requirements are ideally solution neutral; flow-down requirements derived during the solution development phase connect them to a particular solution. Referring to Figure 2 and looking forward, a thorough solution description consists of: (1) a design driven by the requirements and concept of operations, (2) a verification plan traceable to the requirements for acceptance testing the

system, and (3) a validation plan traceable to the concept of operations for commissioning the system.

1	Are the mission need or opportunity, goals, and objectives clear, accurate, and complete?
2	Are the project constraints adequately defined and complete?
3	Are the mission goals and project constraints consistent with each other?
4	Are the system boundary, external interfaces, and key influences accurately and adequately defined?
5	Have the appropriate stakeholders been identified and included in the requirements development process?
6	Is the concept of operations correct and at an appropriate level of detail?
7	Are the requirements clear, accurate, complete, consistent, attainable, and verifiable?
8	Was an appropriate graded-approach used relative to mission importance and project size?

TABLE 2. Suggested review committee charter for a requirements review. A mission review would cover items 1-3, and a stakeholder review items 4-5 and an initial treatment of item 6 [6].

TEN TAKE-AWAY POINTS

1. The majority of the project life cycle cost is committed when turning a conceptual design into a final design, so an up-front investment in requirements and ConOps is important.
2. Engaging key stakeholders provides the domain knowledge needed for establishing an accurate big-picture view.
3. A mission review assesses project-start feasibility by looking for consistency between the goals and constraints.
4. Knowing how a system fits into the rest of the world and how it will be used informs what it needs to be.
5. A stakeholder review assesses early project-readiness by looking for a broad view of the system and engagement of the right people.
6. Organizing requirements by type focuses stakeholder discussions, reveals omissions, and enables re-use between projects.
7. Mapping stakeholders to the requirements and concept of operations identifies whose voice needs to be heard when developing and refining them, and enables focused, efficient, and effective discussions.

8. A requirements review that includes the concept of operations assesses the thoroughness and accuracy of gathering stakeholder expectations and reconciling them with the mission goals, and assesses the project team's readiness to develop a solution that is responsive to the mission.
9. Documenting as you go, especially keeping track of who provided input on what, helps identify missed discussions and establishes an explicit record that provides traceability.
10. A graded-approach is achieved by adjusting the depth of engagement and number of iterations during the process to an appropriate level relative to the mission importance and project size.

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CORRECTIVE MACHINING OF FREEFORM SURFACES USING THE UPC 400

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INTRODUCTION

Freeform optics are increasingly being used in the aerospace, automotive, sensor, LED lighting, and consumer electronics industries for their ability to create more compact and efficient imaging and non-imaging systems [1, 2]. However, the fabrication of freeform optics is challenging as it often requires several independent production steps including data conversions, material processing, and metrology. This multi-step process increases total production time and limits the achievable form accuracy of the optic. A more efficient process allowing for the fabrication and corrective machining of freeform surfaces to a high accuracy could be achieved by integrating the necessary production steps onto a single machining platform. The introduced machine, the UPC 400, provides an integrated platform for fabricating precision freeform optics.

MACHINE DESIGN

The UPC 400 ultra-precision machining center, shown in figure 1, has been designed to efficiently manufacture freeform surfaces up to 400 mm in diameter.

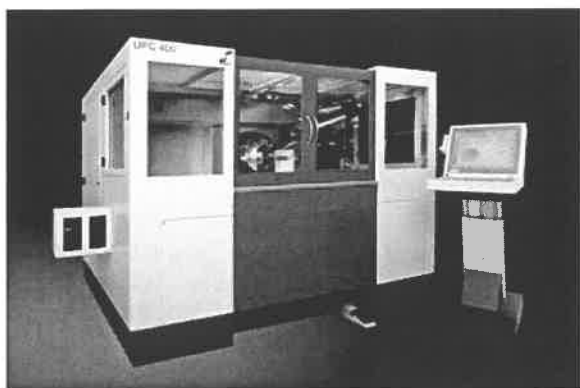


FIGURE 1. The UPC 400 integrates diamond turning, milling, and metrology onto a single machining platform.

Two unique innovations allow the UPC 400 to rapidly machine high precision freeform surfaces. First, all of the necessary machining and metrology processes including diamond turning, milling, and optical measurement are integrated onto the UPC 400 platform. Second, a custom software package provides advanced data handling capabilities for both machining and measurement steps. This combination of hardware as well as software reduces production time, eliminates re-clamping errors, and increases the overall achievable accuracy of the freeform surface.

An Integrated Machining Platform

To enable the machining of large freeform surfaces and to provide a stiff and rigid platform for both diamond turning and metrology, the UPC 400 is built on a solid granite base with hydrostatic guide ways. This base supports the four machine axes (X, Z, C, and B) with high resolution linear scales and spindles which utilize aerostatic bearings to achieve very high accuracies even at fast speeds. This setup minimizes vibrations and increases rigidity, thereby creating an ideal platform for high precision diamond machining and measurement.

A linear X-axis carries the diamond turning equipment with optional fast tool, milling spindle, and metrology system. By arranging all of the tools in a linear setup (i.e. the tools are next to each other on a single X-axis slide) the UPC 400 can quickly switch between process steps without the need to remove or re-clamp the work piece, change setups, or use a rotary axis. Eliminating the need for a rotary axis for tool changes is beneficial since there is greater positioning uncertainty at larger diameters of the rotary table. This uncertainty affects both the tool and metrology position (mainly in X direction) and has a strong influence on the manufacturing accuracy of the tool. The linear arrangement on the UPC 400 reduces these

positioning errors and can be used even for the maximum work piece diameter of 400 mm.

The Z-axis carries the work piece spindle and utilizes a large stroke length for slow tool motion. For freeform machining the spindle also serves as the C-axis. This axis arrangement is advantageous since the slow tool motion is completely independent of the general machine setup. All of the additional tools such as the milling spindle or optional axes such as the B-axis are located on the X-axis slide which is not involved in the dynamic slow tool motion and remains static during the machining process.

The UPC 400 can perform diamond turning using fast tool motion (FTS) as show in figure 2 or a combination of both fast and slow tool motion. The FTS can achieve high acceleration rates and provides a stroke length of greater than 10 mm. This is currently the longest commercially available stroke length for a fast tool setup and provides the ability to quickly machine highly dynamic freeform surfaces without the use of a time consuming slow tool process.

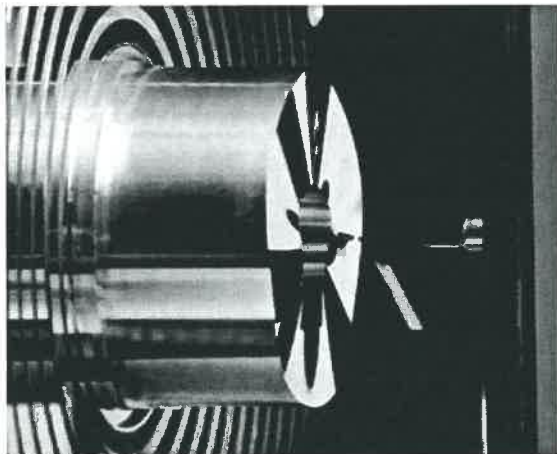


FIGURE 2. Diamond turning a sinusoidal mirror.

The UPC 400 utilizes a unique motion control system which combines an industry standard high power PC-based control with the highest clock rate digital drives for all axes. The data rate for set point generation during freeform machining is significantly faster than today's market standard which leads to higher productivity and better throughput.

The motion control system on the UPC 400 represents a new concept for ultra-precision FTS machining as it uses only one control unit for both the positioning and dynamic axes. This is in contrast to many other commercially

available ultra-precision machines with a FTS in which one set of controls is used for the X, Z, and C axes and a separate controller is used for the FTS.

On systems with separate controllers, the FTS controller calculates a new Z position for the tool based on the current tool position of the X and C axes. Depending on the rotation speed, the lag that occurs during the calculation can cause following errors with respect to the positioning of the C-axis. Furthermore, synchronization issues may occur since the FTS cannot interact with the machine controller. To overcome these issues, the UPC 400 uses a single controller for both the positioning and dynamic axes. This is beneficial as it allows for the fast and slow axes to communicate with each other and remain synchronized during the machining program thus minimizing form error. In addition, the use of a single control unit for all of the machine axes and tools makes the UPC 400 easy for the operator to setup, program, and control.

The X-axis slide features a travel range of greater than 700 mm. This allows for a highly modular machine which can be configured for use with many different processes, tools, and components.

The B-axis on the UPC 400 can be used to increase the machining freedom when using the diamond turning or milling tools. The B-axis is integrated such that the tool is located at the center of rotation. This is beneficial since small angular positioning errors or noise will not affect the radial X position of the tool. Additionally, the Z-axis can move the vacuum chuck to the center of the B-axis, eliminating the need for long projecting adaptors during R/Phi turning. Thus, the rotary B-axis on the UPC 400 can be effectively used without encountering the common errors inherent in standard rotary axis setups.

Besides the manufacturing of mold inserts, the UPC 400 is also well suited for machining large freeform metal optics such as those used in telescopes and other space applications. Hybrid optics, which combines both low and high frequency freeform structures, can also be machined. Alignment features, such as spherical reference marks which can be used to mechanically align the freeform optic during the optical setup, can be precisely machined onto the work piece (e.g. by using the integrated

milling spindle). Since only a single clamping setup is needed for machining all surfaces, these reference marks are accurately placed with respect to the freeform surface. Overall, the availability of different tooling options and mechanical flexibility makes the UPC 400 capable of machining many different types of freeform surfaces during a single process.

Integrated onto the UPC 400 is a 2λ heterodyne interferometer with an extremely high measurement frequency and a resolution of < 5 nm (figure 3). A spiral tool path is used to scan the freeform surface immediately following machining and while it is still clamped inside the UPC 400. This eliminates the need to remove the optic from the machine thereby reducing alignment errors both during the measurement process and when re-clamping the optic for a corrective machining process. Also, the spiral tool path for the measurement utilizes the same surface algorithms as those used for creating the machining tool path so no additional programming steps are needed. The integrated mechanical and software setup coupled with the resolution of the interferometer creates a fast and stable metrology process that is capable of measuring ground or polished surfaces and reflective or transmissive materials.



FIGURE 3. *Measuring a sinusoidal part with the integrated metrology system on the UPC 400.*

Integrating metrology onto the UPC 400 eliminates the need for external measurement equipment, such as a CMM or an interferometer with a CGH which can be both costly and time consuming to use. Working with external metrology equipment such as a CMM can be

challenging as it requires exacting alignment of the freeform surface in order to achieve useful results. Furthermore, the NC-code for the CMM is often generated in a different way than the NC-code used for machining since the CMM software may not directly understand the NURBS data or other description of the freeform surface. Thus, two different data sets are used and a data conversion will be required in order to compare the measured surface against the prescribed, machined surface. This can make it extremely challenging to accurately correct the surface. By integrating the metrology directly onto the UPC 400, the need for data conversions or data corrections is eliminated since all the necessary steps are performed on one platform.

Software Controls

Advanced data handling capabilities power the machining, metrology, and corrective machining processes on the UPC 400. Freeform surfaces are often expressed mathematically as NURBS (Non-Uniform Rational Basis Spline). This format is common to many CAD and optical design programs and allows for the description of complex freeform geometries. After the surface is designed, the NURBS data is converted into a STEP file which provides a lossless format for working with the data on the UPC 400. The STEP file can be used to create the subsequent NC programs for both machining (FTS and milling) and measurement. The ability to directly import the freeform description onto the UPC 400 is beneficial as only a single set of data is needed for all operations (i.e. machining, metrology, and corrective machining programs). This eliminates data conversions and data corrections between process steps for machining and metrology which can distort the data and cause increased surface figure errors.

The UPC 400 can also be directly programmed with G-code generated offline using commercially available software packages. In addition, external measurements of the surface deviation can be uploaded to the machine as a point cloud. The UPC 400 can then calculate a new NURBS surface for corrective machining. In this way, the UPC 400 can be programmed either directly using the onboard software or indirectly using external software programs giving it a range of flexibility and compatibility with other common industry practices.

The software also provides different functionalities to analyze the data. The graphical user interface on the UPC 400 allows for 3-D visualization of the NURBS data and programmed freeform surface. This enables the user to verify that the data has been properly imported and is in the correct orientation and direction (i.e. concave or convex). The software also offers a tool path analysis which analyzes the required axis movements and the clearance angle of the tool to prevent possible collisions.

The initial machining program on the UPC 400 is based on the prescribed NURBS data. Following the initial machining, the freeform surface is measured directly on the UPC 400 using the onboard metrology system. The software can then compare the measured surface to the designed/programmed surface. The resulting error is used to create a new, corrective NC program which can compensate for repeatable errors such as deviations in the tool radius. The need for only a single clamping setup allows for the accurate correction of repeatable tool errors. The combination of both innovative hardware (i.e. single platform machine) and data handling capabilities on the UPC 400 improves the efficiency of the manufacturing process and leads to improved surface accuracies on high precision optical freeform surfaces.

RESULTS

The capabilities of the individual components of the UPC 400 (e.g. slow and fast tool diamond turning, milling, and metrology) have been thoroughly tested. On a 50 mm diameter, spherical aluminum (Al 7075) mirror, the UPC 400 achieved form accuracies better than 60 nm PV (see figure 4) and a surface roughness of < 5 nm Ra. The machining took place with a 0.25 mm tool at 2000 rpm.

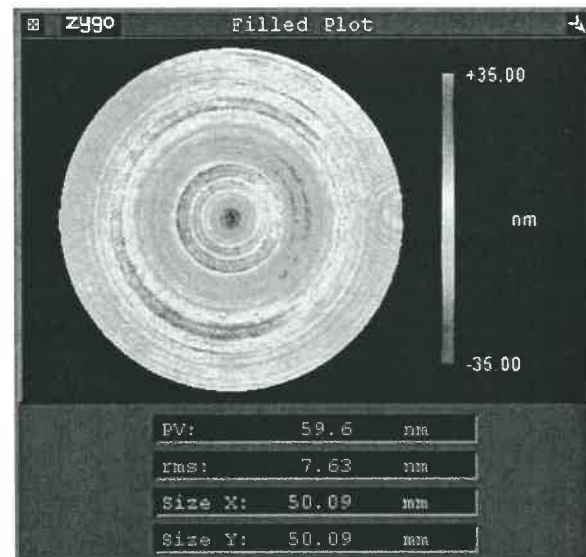


FIGURE 4. On-axis-sphere $R = 100$ mm with form accuracy better than 60 nm PV.

The ability to further improve the surface roughness was limited by the hard intermetallic phases of the aluminum material. Better results of about 1 nm Sa can be achieved using nickel-phosphorous materials as shown in figure 5.

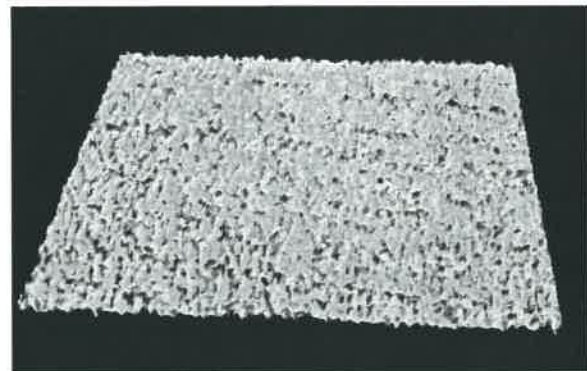


FIGURE 5. Surface roughness measurement on nickel-phosphorous material (50x objective, $S_a = 0.97$ nm).

Freeform Machining Results

In order to test the freeform machining capabilities of the UPC 400, an aluminum spherical optic was machined off-axis. This is the ideal test piece as it requires the full freeform capabilities of the machine and can be quickly and easily measured to a very high level of accuracy using a standard interferometer.

The test part as shown in figure 6, was made of aluminum (AL 6082) with an outer diameter of 200 mm and two opposing, concave spheres located at a radius of 45 mm. Each sphere had a radius of 100 mm and a depth of about 4 mm.



FIGURE 6. Off-axis-sphere for demonstrating the freeform capabilities of the UPC 400.

Machining was performed using a 2 mm radius diamond tool with a spiral distance of 10 μm . The machining time was 2 hours at a speed of 45 rpm.

The results shown in figure 7, demonstrate the ability of the UPC 400 to machine a freeform surface to submicron level accuracy without any corrective machining. The remaining surface figure error is low order with minimal mid-spatial frequency errors. Low frequency error is preferred as mid-spatial errors can significantly affect the performance of an optical system. The higher frequency concentric circles that can be seen in the data are an artifact of the measurement process due to the high reflectivity of the material.



FIGURE 7. Surface measurement of aluminum sphere machined off-axis showing resulting error of less than 650 nm PV.

Further testing using the fast tool system has shown a 3x reduction in process time (compared to machining with the slow tool) while maintaining the same level of form accuracy. Using the fast tool can increase the overall productivity of the machine and significantly reduce machining time. For surfaces with higher acceptable form deviations (e.g. roughing steps), even faster speeds can be achieved.

Corrective Machining

Form errors can be compensated for and corrected using the UPC 400. A measurement of the machined surface is compared against the initial, ideal surface. The deviations between the ideal surface and the existing, machined surface can be used to generate a new NURBS surface for corrective machining.

The top image of figure 8 shows the measured surface of an off-axis machined sphere. The below image of figure 8 shows the new compensated NURBS surface which is based off the initial prescribed surface and measured surface. This compensated NURBS surface contains the necessary information to further improve and correct the surface figure error.

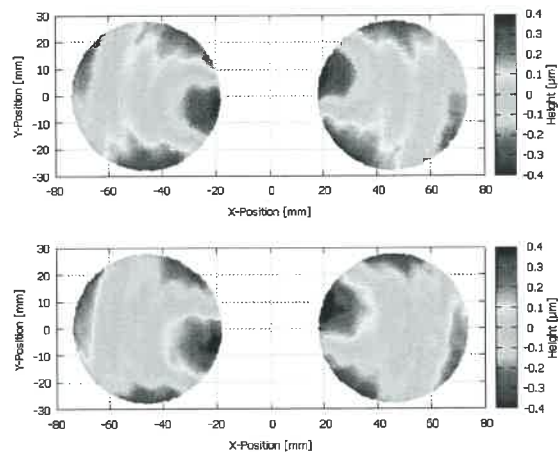


FIGURE 8. *Measured deviation of the spherical lenses after machining (above) and the new NURBS surface for corrective machining (below).*

It is expected that after corrective machining, the surface error will be in the range of 200 nm PV.

CONCLUSION

By integrating fast and slow tool diamond turning, milling, and metrology along with powerful data handling capabilities, the UPC 400 creates an ideal single machine platform for fabricating freeform surfaces. The highest quality form accuracies in freeform machining are achieved in the shortest process time. The single clamping setup preserves the coordinate system enabling the UPC 400 to correct for repeatable machining errors and achieve a high surface accuracy with minimal processing time.

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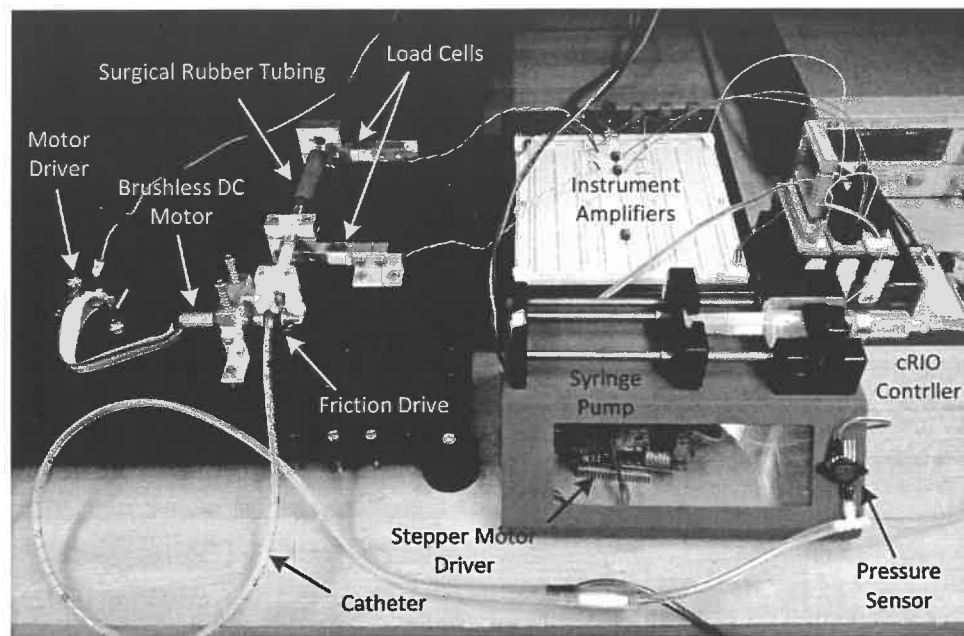


FIGURE 6. Overview of the test bench.

tween the barrel and the plunger. The bougie is connected to a catheter, through which an external syringe pump pushes water into the bougie to modulate the tip extension. A pressure sensor is placed at the outlet of the syringe pump via a T-connector as shown in Figure 6. The sensor measures water pressure in the catheter to estimate the force at the tip of the bougie, which thereby estimates the esophageal stretching force. A friction drive is designed as shown in Figure 5 to feed the catheter back and forth into the esophageal pouch, thereby pushing the closed end of the pouch with the bougie. For more details, see [5].

SYSTEM INTEGRATION AND CONTROL

In the test bench of Figure 6, a mock-up of the esophageal pouch is fabricated out of surgical rubber tubing and mounted on a beam-type load cell to measure the tension during the bougienage. The friction drive is mounted close to the esophageal mock-up to feed the bougie-tipped catheter. A syringe pump is placed next to the bench and connected to the catheter to push water into the magnetic bougie.

The control architecture for the magnetic bougienage system is shown in Figure 7. A cRIO real-time controller, manufactured by National Instruments, is used to implement the control system. The host PC communicates with the cRIO via an Ethernet cable and also via an

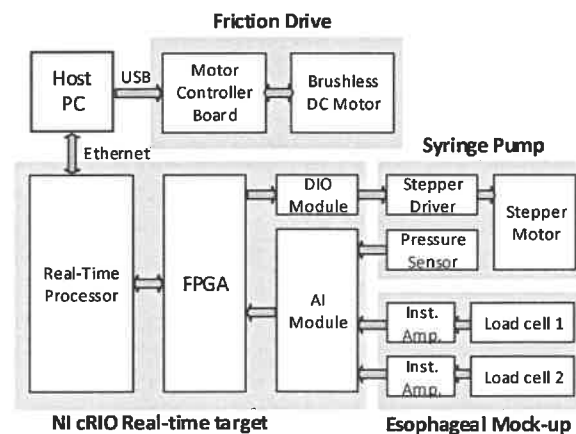


FIGURE 7. The control architecture for the magnetic bougienage system.

USB cable with the EPOS2 brushless DC motor controller board, manufactured by Maxon Motor. The stepper motor in the syringe pump is driven by TB6560 Stepper Motor Driver manufactured by Toshiba.

EXPERIMENTAL RESULTS

We have conducted a bench-level test to demonstrate that the bougienage system can apply a desired stretching force, for example 3 N, to the esophageal mock-up in a controllable manner and can also reliably estimate the stretching force. As shown in Figure 8, we set the neutral gap size of the esophageal mock-up ξ_0 to

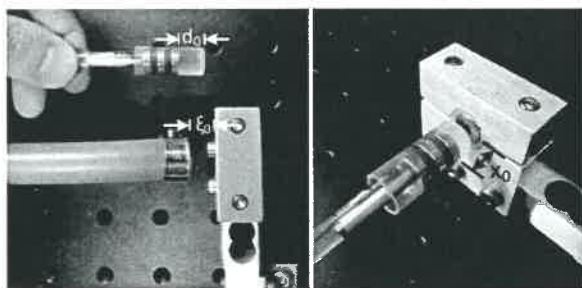


FIGURE 8. The setup for bougienage tests.

10 mm and the tip extension d_0 to 5 mm, which sets the distance between two magnets x_0 to 15 mm. Then, we insert the magnetic bougie into the esophageal mock-up and modulate the displacement of the catheter sinusoidally using the friction drive such that the mock-up experiences maximum tension of 3 N as measured by the load cell. This tension is compared with the bougie tip force estimated with the external pressure sensor. The experimental result is recorded as shown in Figure 9. In this figure, the upper plot is without dither and the lower plot is with dither. Dither is used to average out the O-ring friction of the piston mechanism by adding a 0.4 Hz sine wave to the hydraulic piston fluid drive via the syringe pump. We then filter the pressure sensor output through a moving average filter with a time window of 2.5 sec. In both figures, the red dashed line is the bougie tip force estimated with the pressure sensor after filtering through a moving average filter. The blue solid line is the tension of the esophageal mock-up measured with the associated load cell (the true value). The green solid line is the magnetic force measured with its associated load cell.

In the top trace of Figure 9, we can see that the esophageal mock-up experiences a sinusoidal tension with the maximum of 3 N. However, there is a large discrepancy between the bougie tip force estimated with the pressure sensor and the tension measured with the load cell. It is thus difficult to estimate the mock-up tension from the bougie tip pressure-based force estimate. This discrepancy is due to the sliding O-ring friction between the plunger and barrel, which changes the push/pressure relationship. The bottom of Figure 9 shows the experimental results with dither. We can see that the tension of the esophageal mock-up has small ripples with a frequency of 0.4 Hz caused by the dither. Now, since the dither averages out the friction, the bougie pressure-based tip force estimate matches well with the

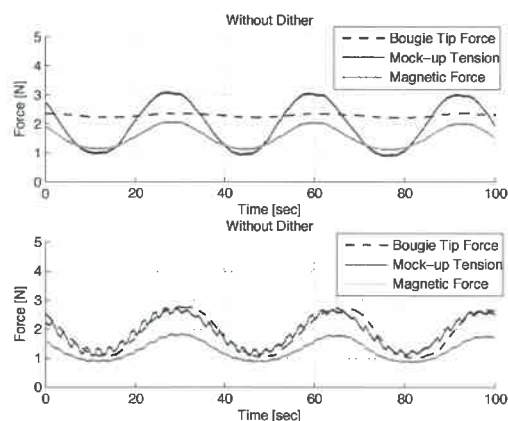


FIGURE 9. Sinusoidal bougienage with $\xi_0 = 10$ mm and $d_0 = 5$ mm. The top trace is without dither, and the bottom trace is with dither.

mock-up tension measurement. Therefore, we can reliably estimate the mock-up tension from the bougie measurement in this case.

ACKNOWLEDGMENTS

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SHORT-DISTANCE REMOTE SENSING USING LOW-COST UAV AND BOARD COMPUTER

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INTRODUCTION

Due to technical improvements of Micro Electro Mechanical Systems (MEMS), high-performance and high-power mechanical parts, such as brushless motor and silicon gyro, are increasingly being used in the consumer market. Unmanned aerial vehicles (UAV), which apply these technologies and are capable of autonomous and automatic flight, are now commercially available for short-distance remote sensing. However, such remote sensing UAVs are still very expensive, and cannot be afforded by small laboratories such as those of universities. Recently, prices of multi-copter UAVs have been reduced to US\$1000 or less on the market; however, these low-cost UAVs can only bear up to sensor weight of several hundred grams, which is not adequate for remote sensing.

On the other hand, the price of embedded computer systems is also dropping. One-board computer systems, such as Arduino and RaspberryPi, are now available at reasonable prices on the consumer market and are increasingly being used for controlling measurement systems and for embedded system education. Especially, RaspberryPi adopts Linux as OS, and has interfaces with various peripherals, such as USB, LAN, and serial ports. Furthermore, RaspberryPi provides options to connect visible and infrared digital cameras [1]. The authors thus propose mounting this board computer to low-cost UAVs to enable use of low-cost UAVs for short-distance remote sensing. Consequently, we developed a near-infrared camera system with RaspberryPi, and demonstrated how data can be acquired with the combination of a low-cost UAV and this camera system.

PLATFORMS AND SENSOR SYSTEM

Low-cost UAV

We chose the DJI Phantom as the low-cost platform product in this research. Phantom is a quad-type helicopter commercialized by DJI and

has the ATT and GPS modes for flight operation [2]. It can hover in the accuracy range of GPS when it receives information from six or more GPS satellites. Figure.1 shows the outlook of Phantom platform. The weight of this UAV is about 900 g including battery, and loading capacity is about 400 g at maximum. The flight time is about 10 minutes without load and decreases as the load increases. We usually set a 5-minutes flight plan in our experiments.

Waypoint Operational UAV

Although the Phantom flight operation is relatively easy, the operator basically have to fly the UAV over the observation targets manually and then take images of the targets. The recent higher functionality UAV has an automatic flight capability by setting waypoints in prior to its flight operation. In order to make a mosaic image or a digital surface model (DSM) for the target efficiently within the limited time period, this waypoint operation should be suitable for our short-distance remote sensing.

Since most of low-cost UAV products do not have the operation mode with waypoints, we tried to assemble our own designed UAV with the use of the flight controller that has a capability to use waypoints. ArduPilot APM [3] version 2.6 is selected as the flight controller for our own assembled UAV. Table 1 shows the major parts list of our UAV, and Figure 2 gives the outlook of our waypoint capable UAV. The weight of the UAV is about 1050 g including battery, and the loading capacity is about 500 g at maximum.

Near-infrared GPS Camera System

Using RaspberryPi as the controller, an imaging system for acquiring aerial photographs was designed to satisfy the following conditions:

- (1) Total weight of about 300g including the batteries.
- (2) Input power of 5V DC.
- (3) Operational time of more than 10 minutes.

Figure 3 shows the outlook of our camera system. We connect the GPS as well as near-infrared camera modules to the I/O of RaspberryPi, and use the mobile battery for portable phones as the power source (5V 1A · 2700mAh). The weight of the developed system including the battery is 231.3g and operational time period about 110 minutes. We attach the developed system to the UAV using a simple anti-vibration mount.

We also developed a control program for this camera system applying multi-threaded processing technique in order to acquire data simultaneously from the camera and the GPS modules. This program is automatically activated when the system power is turned on. This control program starts to take pictures with selected time intervals, currently set to 2 seconds, and records GPS logs once the GPS modules can calculate the location coordinates with better accuracy. The camera system can record the shooting time, latitude, longitude, and altitude in the Exif area of the image file. It finishes shooting and recording of the GPS logs using external commands. Figure 4 gives a sequence diagram of our camera system in UML format.



FIGURE 1. Low-cost UAV used in this research (DJI Phantom).

TABLE 1. List of assembled UAV parts

Controller	APM 2.6
Motor	880KV brushless motor X4
Propeller	10 x 4.5 propeller X4
Battery	3S 11.1V 2200mAh LiPo battery
Sensors, etc.	GPS + compass receiver
	6 ch radio receiver
	battery monitor



FIGURE 2. Outlook of assembled UAV with waypoint operation.

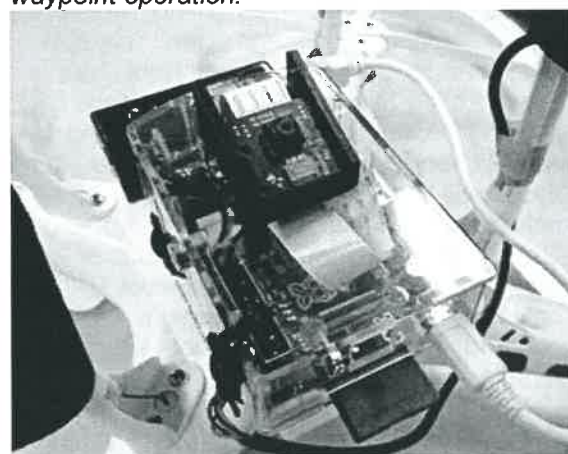


FIGURE 3. Near infrared camera system with Raspberry Pi.

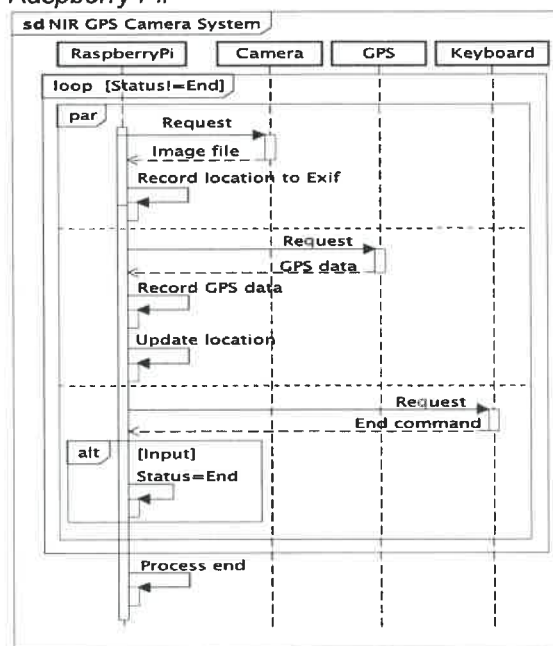


FIGURE 4. Near infrared camera system with Raspberry Pi.

ACQUISITIONS AND PROCESSING OF TEST DATA

During the period from March 7 to March 10, 2014, we had an opportunity to visit the tsunami-affected areas in Miyagi prefecture. Test data were acquired by using the Phantom UAV with our near-infrared camera combined with a superblue filter [4] that eliminates the red band. Figure 5 gives an example of the acquired data over the area that was washed away by tsunami in Ishinomaki city colorized in NGB (assigned Near-infrared band to red channel, Green band to green channel, and Blue band to blue channel) as well as Normalized Difference Vegetation Index (NDVI) image.

The recorded image data has an Exif area including latitude, longitude, and altitude information obtained from the GPS receiver. These camera locations were used as the initial values for more accurate location estimation, and then orthophoto and DSM mosaic images were created using the structure from motion (sfm) technique. Figure 6 gives an example of these processes. Figure 6(a) shows each individual image, and Figure 6(b) and 6(c) indicate mosaicked ortho and DSM images, respectively. These data were taken at the embankment model at the time of reconstruction in Yuriage district of Sendai city.

SUMMARY AND FUTURE RESEARCH PLANS

We developed a short-distance remote sensing system with the use of a low-cost UAV and board computer. By using the near-infrared camera controlled by the board computer, we confirmed that the developed system could be effectively used for our short-distance remote sensing purpose. Our future research plans are as follows,

- (1) Real time monitoring of flight information using XBee communication.
- (2) Reducing the weight of the camera system.
- (3) Simultaneous observation by using 4 bands including blue, green, red, and near infrared bands.
- (4) Calibration of output image from camera system.

We also hope to improve the accuracy of real-time GPS locations and achieve effective data acquisition using multiple platforms.

ACKNOWLEDGEMENTS

The test data used in this paper was obtained with the assistance of Professor Kohei Cho of

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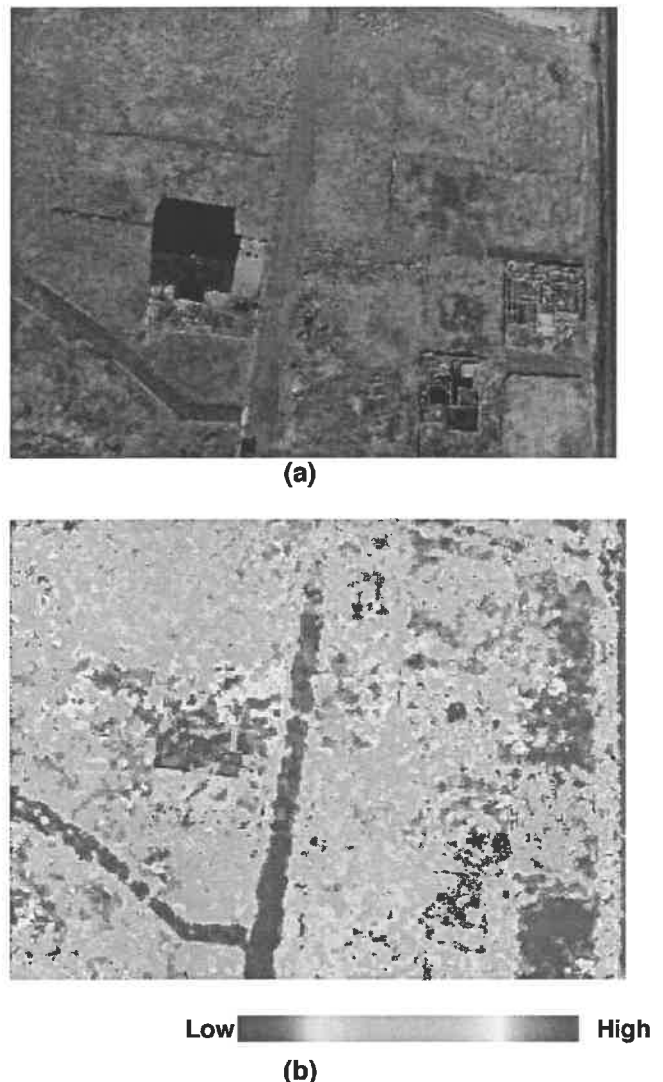


FIGURE 5. Example of NGB and NDVI images created from data acquired in Ishinomaki. (a) NGB color image and (b) NDVI image for same area of (a).

OVERVIEW OF THE SYSTEM

A schematic and a photo of the proposed motion stage are presented in Figure 1. The moving body consists of a monolithic shaft with plates attached to planar grooves machined on the top and bottom sides. The structure is supported by two air bushings at each end, thus constraining lateral, vertical, pitch, and yaw motion. The flat air bearing on the bottom constrains roll motion. The bearing structure is very easy to assemble and align around the ground cylindrical shaft. The two ends of the shaft are attached to magnetic cores, which engage into stationary coils fixed to brackets, thus constituting VCMs in moving magnet actuator (MMA) configuration. The two magnetic cores help to compensate for the nonlinearity caused by the change in each other's engagement. Position measurements are obtained through a sinusoidal encoder, with the head mounted in a stationary manner. Thus, there is no contact with the moving section. There is also provision in the design to incorporate acceleration feedback, or swap the flat air bearing with a vacuum preloaded air bearing for additional damping.

DESIGN AND OPTIMIZATION OF THE VOICE COIL MOTORS (VCM)

A schematic of the VCM is given together with an example set of COMSOL FEA results, in Figure 2. The VCM is made up of the AISI 1018 iron core and the wound coil which engages to the core within a specified clearance. An axially magnetized, NdFeB ring magnet with Nd42 grade has been used as the flux source. In the design of the VCMs, two sets of optimizations have been carried out. The first set charts out the performance of the system in terms of "acceleration per current density", which penalizes added mass by the magnetic core while seeking the maximum force achievable by the current carrying capacity of the coil wires. The second optimization index is the "motor constant" formulated for the volume of coil, which is a measure of square root of heat dissipated per unit force, independent of the coil design. For each set of optimizations, the resulting gap flux densities in the radial direction of the coil were estimated by simulating the corresponding VCM design in COMSOL, as exemplified in Figure 2. By maximizing the first performance factor and minimizing the latter, a design which allows high positioning performance can be achieved with minimal thermal distortion caused by the motors.

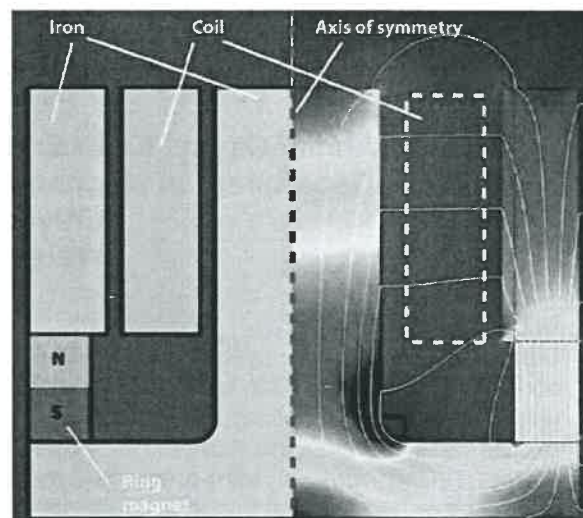


FIGURE 2. Schematic of VCM components with FEA results for one design point.

VIBRATION MODES

An analytical study of the vibration modes using catalog values for air bearing/bushing stiffnesses [6] has been carried out. The diagram which gives the definitions for vibration axes for first natural frequencies is given in Figure 3.

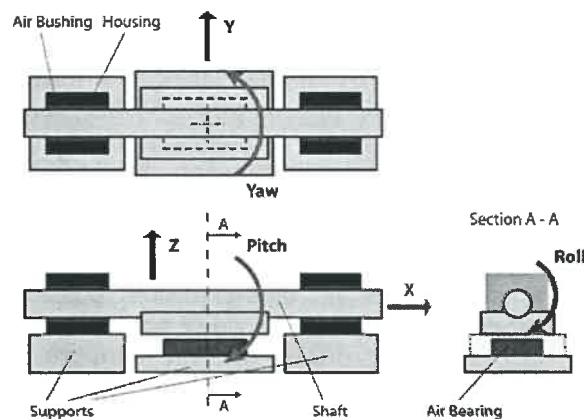


FIGURE 3. Vibration axis definitions for first natural frequencies.

First natural frequencies are calculated as, 940 [Hz], 940 [Hz], 736 [Hz], 735 [Hz], 475 [Hz] for Y, Z, Yaw, Pitch and Roll axes, respectively. The roll mode is calculated using an approximate approach by integrating bits of linear stiffness elements over the area under compression. The accuracy of these predictions is to be confirmed with modal testing.

CONTROL SYSTEM DESIGN

Current Control Loop

The current control loop was realized using linear amplifiers, tuned at 1000 [Hz] bandwidth. The current controller is made up of an integrator and a lead. The same controller is implemented for both motors using separate amplifiers. Motor properties (in terms of inductance) change depending on the position of the stage, as this changes the level of engagement for each motor. Therefore, closed loop transfer function (CLTF) of the current control loop varies with stage position. In order to validate the implemented controller, the stage is clamped at various stroke positions, and CLTF is evaluated using sinusoidal voltage commands sent to the amplifiers. The resulting frequency current response is recorded. Results are presented in Figures 4 and 5. The behavior of Amplifier A + Motor A and Amplifier B + Motor B, in terms of the peak magnitude and phase roll-off, follow the same patterns in reverse order with respect to the stage position. Theoretical prediction for current control CLTFs at $x = 0$ [mm] and $x = 20$ [mm] are also presented in Figures 4 and 5. These are evaluated using the initial motor identification results together with the designed current controller. Magnitude plots show rather close agreement in both cases, while the phase plots seem to have a sharper decline in the experimental case. These are thought to have originated from discrepancies in the initial identification of the motors.

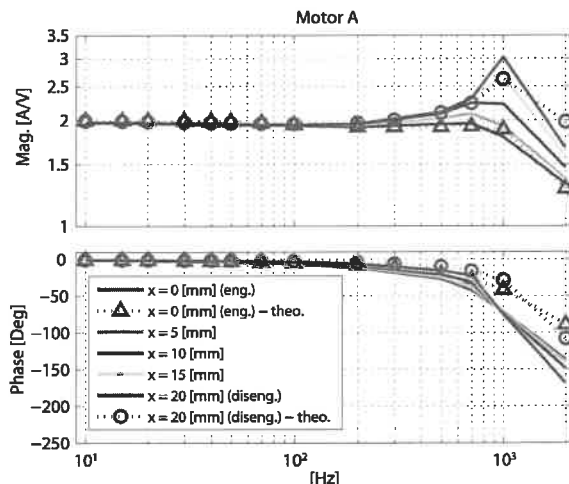


FIGURE 4. Bode plot for voltage to current transfer function of Motor A for various fixed positions of the stage.

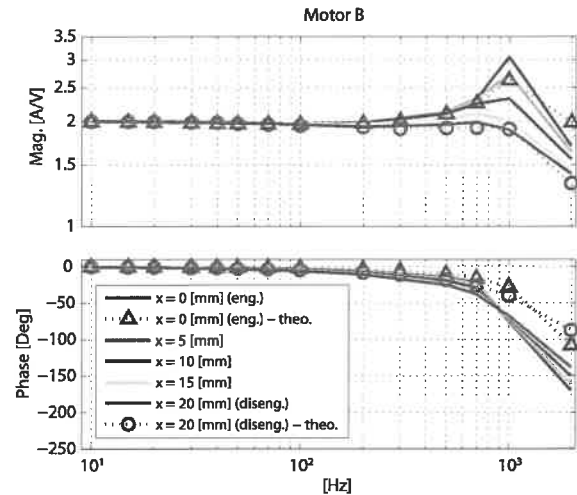


FIGURE 5. Bode plot for voltage to current transfer function of Motor B for various fixed positions of the stage.

Position Control Loop

The experimentally obtained open-loop position frequency response of the motion system to the amplifier input voltage is given in Figure 6.

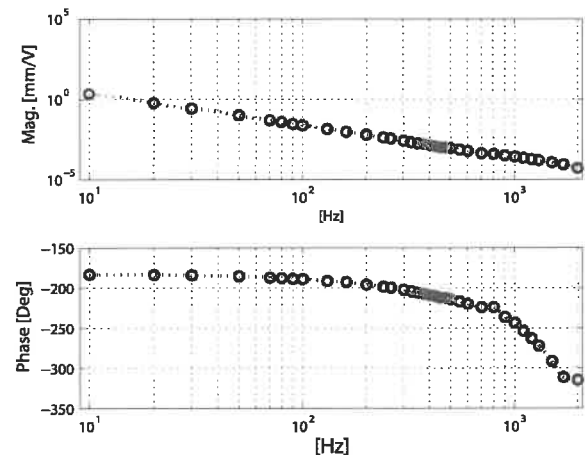


FIGURE 6. Open loop position response.

In preliminary testing, the position control loop was designed as a lead-lag controller, tuned for 500 [Hz] cross-over frequency. The controller was implemented on a dSPACE - DS1005 board sampling at 20 [kHz]. The loop transfer function, $L(s)$ is given in Figure 7. The upper plot represents the Bode magnitude plot, while the lower one is the Nyquist diagram of $L(s)$ near the origin. A phase margin of $PM = 40^\circ$ was used.

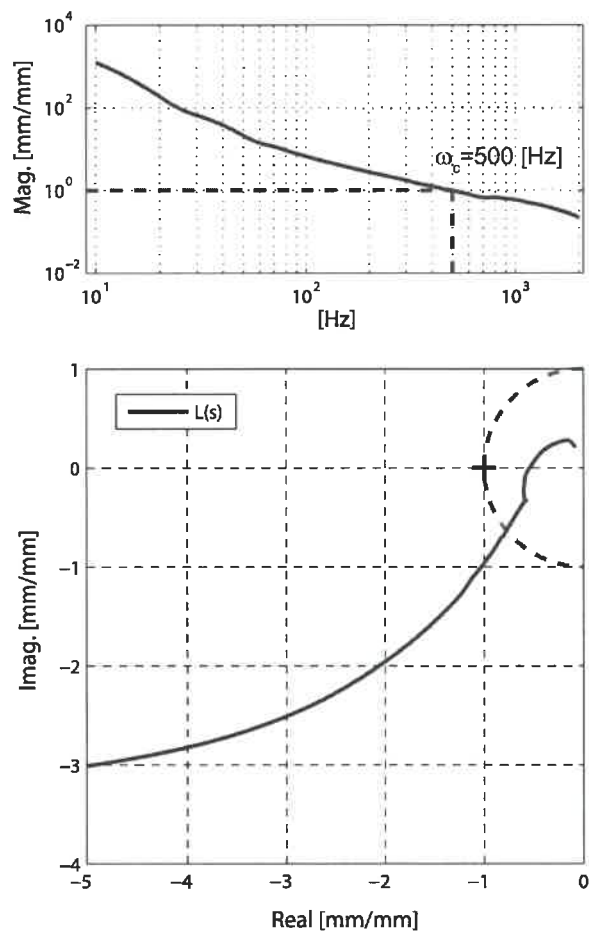


FIGURE 7. Bode magnitude plot (upper) and Nyquist diagram near the origin (lower) for $L(s)$.

CONTROL PERFORMANCE EVALUATION

Initial positioning tests, with filtered step inputs (cut-off freq. 40 [Hz]), demonstrate a settling resolution of 5 [nm], as shown in Figure 8.

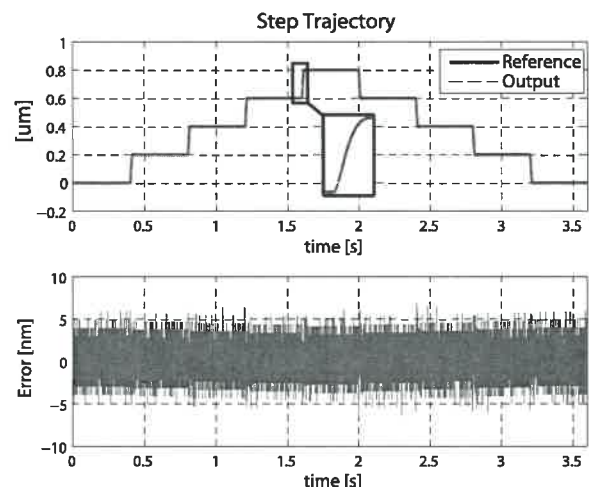


FIGURE 8. Tracking performance for filtered step trajectory.

A separate study has been carried out to gauge sensor noise. The stage was mechanically fixed to its position at the middle of the stroke length and position recordings at 20 [kHz] were gathered from the sensors reset to zero at the beginning of the recording. A sample from these measurements, all of which reflect similar characteristics, is presented in Figure 9.

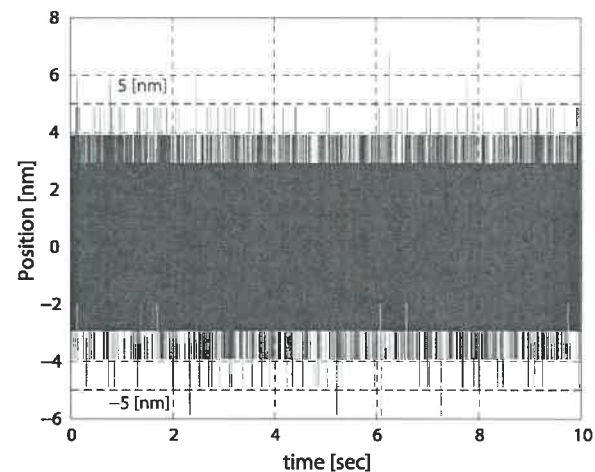


FIGURE 9. Position signal recorded at the clamped position at the middle of the stroke length.

The signals recorded should be a combination of vibratory motions of the reading head with respect to the fixed stage, and the measurement noise. FFT of the sample signal in Figure 9 up to the cut-off frequency 10 [kHz] is given in Figure 10.

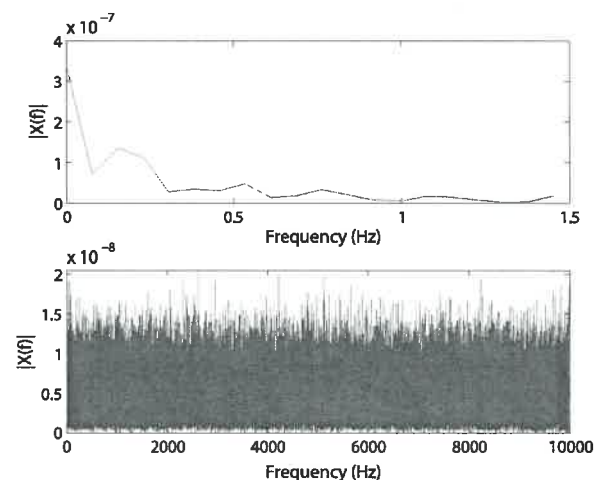


FIGURE 10. FFT results for position signal recorded at clamped position.

Results in Figure 10 indicate that the recorded position signal is a combination of drift at very low frequency and uniformly distributed high frequency noise. Hence, the measurement noise floor is established at 10 [nm] peak-to-peak, which coincides with the positioning resolution. It can be concluded that the positioning resolution of the system is currently limited by the sensor characteristics.

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ANALYSIS OF MATERIAL MECHANICS FOR PNEUMATIC RUBBER ACTUATORS (MATERIAL PROPERTIES EXPERIMENT AND MOTIONS OF NBR BELLOWS ACTUATORS)

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INTRODUCTION

The pneumatic actuator made from rubber has the flexible feature. This is also called a soft actuator in order to carry out flexible operation. A soft actuator has the soft and safe feature. The actuator can change also corresponding to external force. Therefore, this actuator is excellent in the machine attached to a human body and in-pipe mobile robot. This actuator is studied for the power assistant suits which assist elderly people, a physically handicapped person, and the worker that carries a heavy load [1].

The photograph of a pneumatic rubber bellows actuator is shown in Fig. 1. We use these for the in-pipe mobile robot which shows in Fig. 2. The pneumatic rubber bellows actuator is excellent in the power-weight ratio, and even if small, it can take out large generative force. Therefore, the actuator is used for the in-pipe mobile robot which can move freely in the inside of a small diameter pipes [2].

Only the power which a design of the power assistant suit and in-pipe mobile robot using a pneumatic rubber actuator produces by air pressure was examined. The characteristic peculiar to rubber which arises by rubber has been disregarded. The simulation cannot finish catching the motion of an actuator and is designed by the Trial & Error technique. In this research, the rubber which constitutes the actuator used for the in-pipe mobile robot was

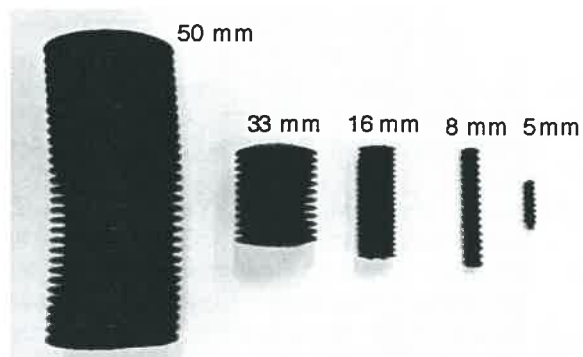


FIGURE 1. NBR bellows actuators

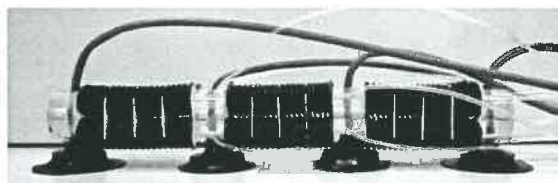


FIGURE 2. Green Caterpillar type In-pipe mobile robot

analyzed experimentally. We thought that a power assistant suit and a robot could be designed theoretically, when the action of the actuator had been grasped. Rubber carries out nonlinear large deformation by external force. The material characteristics of rubber were identified by experiment and the database was built. This can expect improvement in

performance and reduction of development costs

MEASUREMENT OF THE SHORE-A HARDNESS OF NBR

The hardness which is the material characteristics of rubber is measured. We are fabricated the actuator using the rubber called NBR 60, NBR 70, NBR 80, NBR 85, and NBR 90. Based on JIS K 6253, the Shore A hardness of various rubbers was measured using the type A durometer. Measurement piled up three 300mm * 300mm * 3 mm rubber sheets. The hardness measurement result of NBR is shown in Fig. 3. It turned out that a little Shore A hardness of the numerical value of a name and measured value differs. We think that the Shore A hardness rougher than a declared numerical value can be grasped.

THE TENSILE TEST OF NBR

The tensile test was done in order to search for the tensile strength which is a material constant of NBR. The tensile test used the dumbbell-like No. 3 specimen which is a calibration block based on JIS-K-6251. The tensile strength to cutting and the growth of cutting were measured at the tension speed of 500 mm/min. A tensile strength σ_B (MPa) is calculate by the Eq. 1.

$$\sigma_B = \frac{F_B}{A} \quad (1)$$

F_B (N) shows maximum tensile load and A_0 shows the cross-sectional area of the gage line of a specimen. Measurement of the maximum tensile force F_B (N) is inputted into PC using a digital recorder. The growth at the time of cutting is calculated by the Eq. 2.

$$\varepsilon_B = \frac{L_1 - L_0}{L_0} \quad (2)$$

L_0 shows the distance between gage lines and L_1 shows the distance between gage lines at the time of cutting. The distance between gage lines was measured by picture judging. By doing a tensile test, the stress distortion diagram was able to be obtained about five kinds of NBR which we are using actuators. As an example, the nominal stress-nominal value distortion diagram of NBR 70 is shown in Fig. 4. A result shows at NBR that nonlinear modification has arisen. The growth at the time of cutting is shown in Fig. 5. When rubber becomes hard, it turnsout that growth decreases.

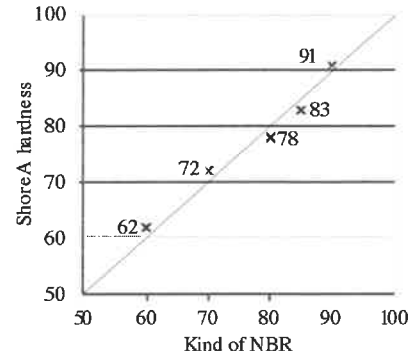


FIGURE 3. Measurement of shore A hardness

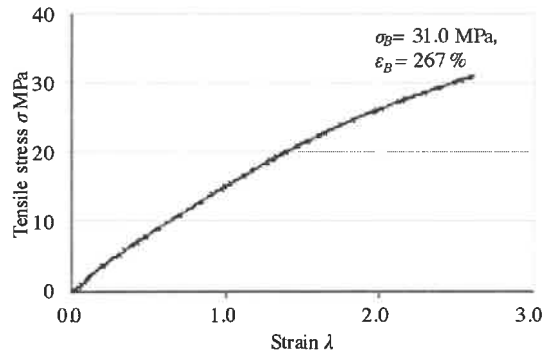


FIGURE 4. Stress-strain curve of NBR 70

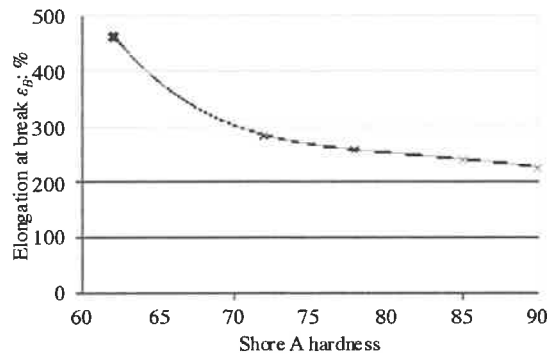


FIGURE 5. Elongation at break

NBR 60 is understood that growth is comparatively large. This shows that the growth of the material of rubber is not hardness and proportionality relation.

STRAIN ENERGY FUNCTION EXPRESSION OF NBR

FEM analysis of the nonlinear deformation of rubber is conducted. The Mooney-Rivlin type was used for this analysis. Expression modeling

of the specimen of rubber was carried out. The strain energy function of NBR was drawn from there. The stress σ concerning a rectangular section was denoted by the extension ratio λ . The primary model of a strain energy function is denoted by Eq. 3.

$$\sigma = C_1(2\lambda - \frac{2}{\lambda^2}) + C_2(2 - \frac{2}{\lambda^3}) \quad (3)$$

We applied the tensile test data of NBR 70 shown in Fig. 4 to the Eq. 3 and the least-squares method obtained the coefficients C_1 and C_2 . As a result, it became a value of $C_1=0.0966$ and $C_2=0.0320$. Verification of the stress-distortion diagram of a primary model is shown in Fig. 6. x in a Fig. 6 expresses measured value and the solid line expresses strain energy function expression. It was judged that agreement of measured value and strain energy function expression was not enough. We found being improved of the model from the result. The secondary model of the strain energy function was examined. This is denoted by Eq. 4.

$$\sigma = (2(C_1 - 6C_2) + \frac{2(4C_2 + C_3)}{\lambda} + 4C_3\lambda^2)(\lambda - \frac{1}{\lambda^2}) \quad (4)$$

The tensile test data of NBR70 is similarly applied to Eq. 4, and the coefficients C_1 , C_2 , and C_3 are identified with a least-squares method. The result was set to $C_1=0.491$, $C_2=14.6$, and $C_3=0.617$. Verification of the stress-distortion diagram of a secondary model is shown in Fig. 7. Measured value and strain energy function expression are well in agreement. Therefore, it was judged that a secondary model was suitable for derivation of a strain energy function.

CONCLUSIONS

The material-characteristics experiment was conducted on NBR which is the material of the a pneumatic rubber actuator used for a power assistant suit and the in-pipe inspection robot.

The primary model examined the strain energy function using the Mooney-Rivlin type. The tensile test result of NBR70 was applied to the formula, and the coefficient was obtained. However, the agreement with measured value was not accepted.

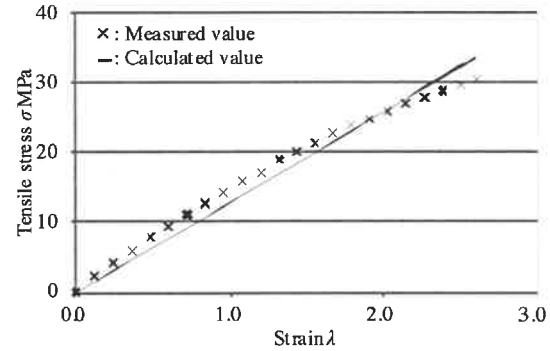


FIGURE 6. Verification of the primary model

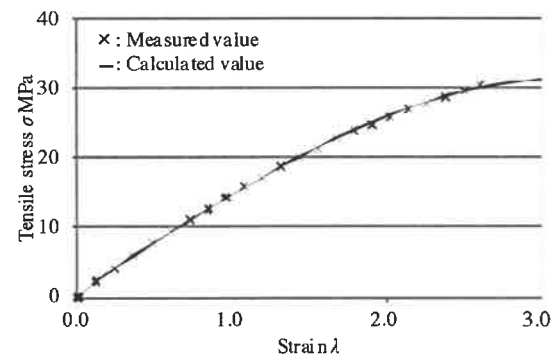


FIGURE 7. VERIFICATION OF THE SECONDLY MODEL

The secondary model examined the strain energy function for the improvement of a model. With the application of the tensile test result of NBR70, the coefficient was obtained like the primary model. As a result, since agreement with measured value was accepted, it can be judged that a secondary model is suitable for a strain energy function.

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A ROLL-BONDING MACHINE FOR POLYMERIC FILMS

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MOTIVATION

An ongoing activity at the NVS-MIT Center for Continuous Manufacturing (NVS-MITCCM) envisions continuous manufacturing of pharmaceutical products from thin polymeric films. Such a manufacturing process is intended to operate in a minimum number of steps, and the final drug-product be made from start to finish in a continuous manner at one facility. This methodology promises to accelerate the introduction of new drugs in the market, minimize the generated waste, reduce the use of energy and raw materials, carry out quality checks inline, as opposed to post-production, and increase the overall reliability and flexibility of the production process [3]. A crucial step in this downstream process is bonding of multiple incoming layers of films (each layer $\sim 100 \mu\text{m}$ thick) in a continuous mode. Here, we present the design of a continuous roll-bonding machine to carry this task with precision.

Notation

h_1	film-stack thickness at the inlet [m]
P	line loading [N/m]
h_o	total thickness at the outlet [m]
e^p	plastic strain in film-stack [m/m]
L	total load [N]
w	width of the film [m]
M	moment per unit width [N]
E_{film}	modulus of film [MPa]
$\sigma_{y,film}$	yield strength of film [MPa]
R	radius of roller [m]
E_{roller}	modulus of roller [MPa]
$\sigma_{y,roller}$	yield strength of roller [MPa]
τ	duration of contact in rollers [s]
V_2	exit speed of the films [m/s]
$\bar{\sigma}_x$	average principal stress in x-direction
$\bar{\sigma}_y$	average principal stress in y-direction

FUNCTIONAL REQUIREMENTS OF MACHINE

Bonding multiple incoming layers of polymeric films during continuous manufacturing requires

appropriate levels of active plastic deformation over a certain interval of time [2]. The polymer films of interest have a yield strength $\sigma_{y,film} \sim 6-8 \text{ MPa}$ and a modulus $E_{film} \sim 200-300 \text{ MPa}$, and feed speeds around 30 mm/min . Typically 10%-20% plastic deformation is required over a short interval ($5-20 \text{ s}$) to achieve bonding. The machine should have sufficient flexibility to operate in a wide range about these specifications.

DESIGN AND ANALYSIS OF ROLL-BONDING MACHINE

A possible solution is to design a machine comprising of two rigid-rollers capable of exerting sufficient loads over the desired interval of time. Rollers made of stainless steel (yield strength, $\sigma_{y,roller} \sim 600 \text{ MPa}$ and a modulus $E_{roller} \sim 180-200 \text{ GPa}$) can be treated rigid in comparison to relatively ductile polymer films. Flattening of rollers, which may be an issue in the context of rolling metals, is unlikely here since polymeric films have a small yield strength in comparison with that of the rollers. The steel rollers can also provide sufficient traction on the incoming films for necessary forward drive.

In general, the total strain (e^{total}) comprises both elastic (e^e , recoverable) and plastic (e^p , non-recoverable) strains, i.e. $e^{total} = e^e + e^p$. To a first approximation we can ignore the elastic strains assuming that the plastic strains will dominate in this deformation processing situation, i.e. $e^{total} = e^p$. Therefore the material can be treated as rigid-plastic, i.e. a material which is perfectly rigid prior to yielding and perfectly plastic afterwards. The analysis presented here is adopted from [1]. A rigid perfectly-plastic rolling scheme is shown in Figure 1, and from the rolling geometry we have

$$d = \frac{a^2}{R} \quad (1)$$

where,

$$d = \frac{h_1 - h_o}{2} \quad (2)$$

The von Mises yield criterion in conjunction with plane-strain condition (in the y-direction) leads to

$$|\bar{\sigma}_x - \bar{\sigma}_z| = \frac{2}{\sqrt{3}} \sigma_{y, film} \quad (3)$$

By considering the equilibrium of a differential element and following the von Karman's procedure [5], the line loading (P) and the moment per unit width (M) applied to the rolls can be estimated as follows:

$$\frac{P}{(\sigma_{y, film} / \sqrt{3})a} = 2 + \frac{a}{\bar{h}} \left(\frac{1}{2} - \frac{1}{3} \frac{a}{R} \right) \quad (4)$$

$$\frac{M}{(\sigma_{y, film} / \sqrt{3})a^2} = 1 + \frac{a}{4\bar{h}} \left(1 - \frac{a}{R} \right) \quad (5)$$

where, $\bar{h} = \frac{h_1 + h_o}{2}$ is the mean film thickness. From the geometry of deformation, if V_2 is the exit velocity of rolled-stock, the time of compression (τ) in the rollers (if $d \ll R$) can be estimated as

$$\tau = \frac{\sqrt{R(h_1 - h_o)}}{V_2} \quad (6)$$

Typically if the coefficient of friction between the rollers and the strip is large, and/or the strip has lower yield strength the frictional traction at the interface exceeds the yield stress of the strip in shear so that there is no slip in the conventional sense at the surface i.e. plastic shear will take place in the rolled stock, while the surface will "stick" to the rolls with static friction. It is worth emphasizing that the above procedure incorporates no-slip assumption and, homogeneous deformation i.e. the vertical segments of the bar deform vertically as if they were separated from each other so that no shear stress can arise in them. After considering several trade-off designs we selected steel rollers of $R = 100 \text{ mm}$, so as to achieve desired active plastic-straining during rolling. For example, $\sigma_{y, film} = 6.0 \text{ MPa}$, $h_1 = 1 \text{ mm}$, $h_o = 0.8 \text{ mm}$, then $2d = 0.2 \text{ mm}$ (indicating 20% plastic compression), and $a = 4.47 \text{ mm}$.

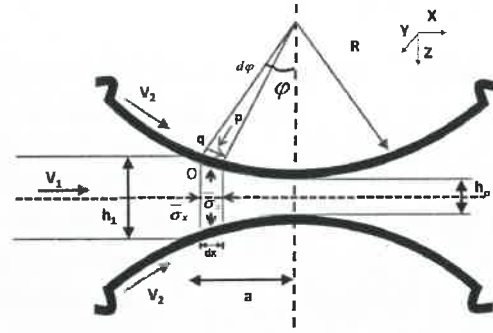


FIGURE 1. Rigid perfectly-plastic rolling scheme.

From equations 4 and 5, $P = 6.82 \times 10^4 \text{ N/m}$ and $M = 150.89 \text{ N}$, respectively. If we assume the width of the strip to be 20 mm , then the total load $L = 1359.3 \text{ N}$ and total torque is 3.01 Nm . If V_2 is 30 mm/min then the residence time (τ) would be 8.94 s . This illustration demonstrates that we can successfully achieve active plastic-deformation in the rollers in a few seconds.

It is worth mentioning that rigid-plastic analysis presented here does not take into account any strain hardening, heating effects, etc. and in actual process compression loads can differ. Other machine elements, discussed next, were sized and selected appropriately for successful functioning of the roll-bonding machine. For the sake of brevity we abstain from presenting those calculations here. In principle, sophisticated finite element procedures along-with accurate constitutive material models can be employed for accurate predictions of stresses, strains, etc.; however, the goal of this paper is to present analyses to identify the approximate operating loads and thereby select appropriate machine elements for a roll-bonding machine. During actual operation, like any other operating machine, the compression loads and angular-speed of rollers are tuned so as to meet the desired performance.

Figure 2 shows the CAD model of roll-bonding machine that we have developed. Table 1 lists the main components of the machine along with their functionality. The two cylindrical rollers with shafts are mounted on two separate 'U' shaped stages. The stages carrying the rollers are attached on the linear-slides such that distance between them can be adjusted. The distance between the rollers is measured by a micrometer. The position of the roller-carrying-stages attached to the linear slides is set using the two

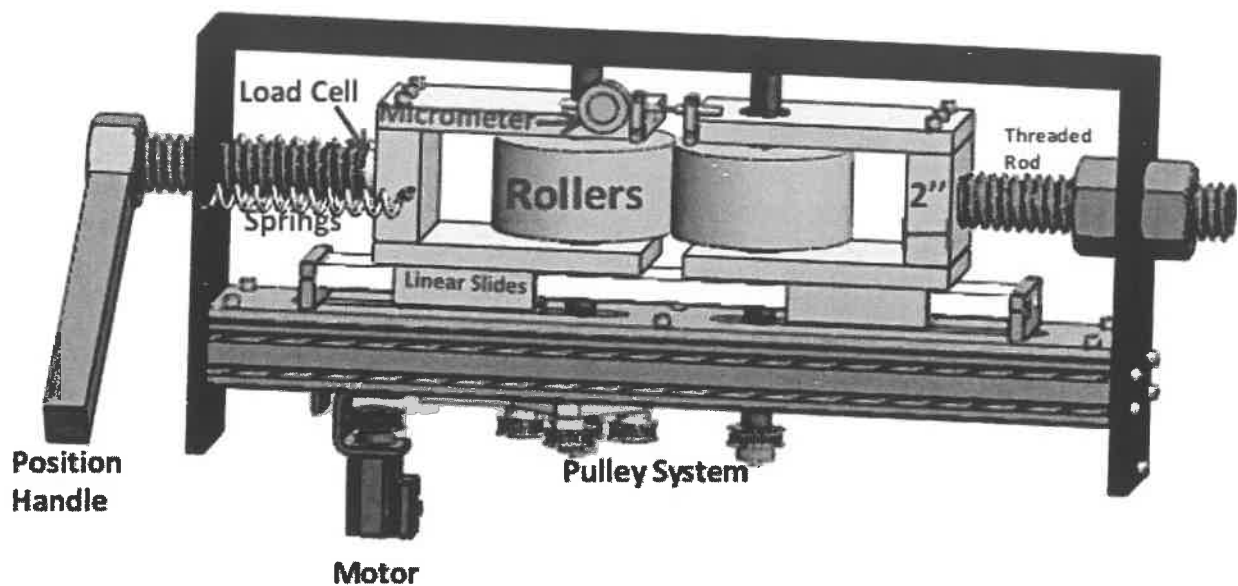


FIGURE 2. CAD model of the Roll-Bonding Machine. The cylindrical rollers provide adequate line-loading to achieve plastic deformation on incoming films over a desired interval of time.

TABLE 1. Major components of the Roll-Bonding machine and their functionality. Functional requirements of each element have been carefully analyzed ([4]) before making design choices. Machining and assembly were carried out such that the parallel between the rollers was accurate up-to $\pm 0.003''$.

Machine Element	Functionality
Rollers	Compression rollers capable of applying several kilo-Newtons of line-load were machined out of stainless steel with a surface finish of $R_a \sim 6\mu m$. The diameter of each roller is 200 mm.
Linear Slides	Carry the stage of the compression rollers, and enable setting the relative gap between the rollers. The slides were purchased from Stelron Components Inc.
Micrometer	Measures the distance between compression rollers which are mounted on the linear slides.
Load Cell	Purchased from Omega (model LC 305-1K) to measure the compression load (N). It is mounted on the left roller-stage.
Springs	Extension springs are used to secure the left roller-stage and load cell against the position handle.
Position Handle	Drive the left roller-stage through a threaded-rod and set its position.
Threaded Rod (Right)	Set the position of the right roller. Locking nuts secure it in the desired position.
Pulley System	Attached to the shafts of the rollers, and interconnected by a double-sided timing belt (not show so as to avoid clutter).
Stepper Motor	Purchased from Applied Motion Products (Model STM-23 QN). An appropriate gear train was selected to provide a minimum angular speed of 0.08 rpm and maximum torque of 50 Nm to drive the compression rollers.

threaded-rods. A load cell is mounted on the left roller-carrying-stage to measure the compression loads. The two threaded rods are secured to the main frame by the action of locking-nuts. During operation the position of the right-roller-stage is kept fixed, and the position of the left roller-stage is adjusted through threaded rods and a position handle. The rotatory motion of the position handle that drives the left threaded rod causes the sliding motion of its roller-stage. Two extensional springs are mounted on the frame and the left roller-carrying-stage such that load cell is well secured against the left threaded rod.

During operation, the position of the left roller-carrying stage is adjusted till a desired level of load or displacement is noted on the load cell or micrometer, respectively. The shafts are driven by a system of pulleys and double-sided timing belts using a stepper-motor. The stepper motor is attached to the main frame of the machine, and during operation an appropriate level of tension is maintained in the timing belt which drives the pulleys on the shafts. To ensure that timing belt does not slip over the pulleys when driving the shafts, an idler pulley is mounted to take up the slack in the timing belt, which occurs as rollers are brought closer. The angular speed of the motor dictates the exit speed (V_2) of the films from the rollers. Figure 3 shows a snapshot of roll-bonding of multiple layers through our devised mechanism.

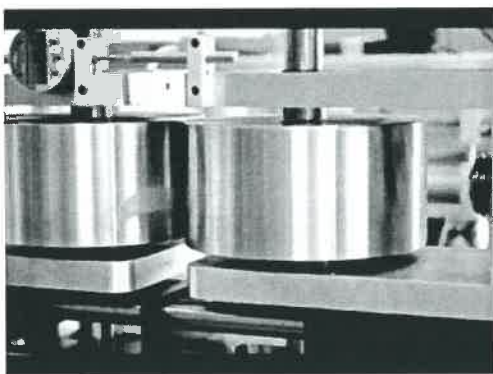


FIGURE 3. Roll-bonding of polymeric films.

The overall success of the roll-bonding mechanism lies in precisely applying the line-load without introducing unwanted deflection or failure of any machine element during compression between the rollers. In order to evaluate the accuracy of our system we have tested the stand-by performance (i.e. compression without the presence of films) in the displacement-controlled

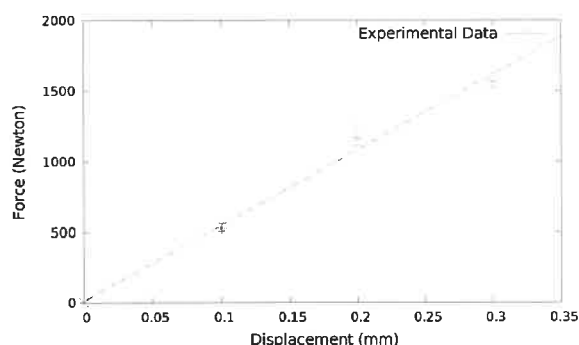


FIGURE 4. Load vs Displacement characteristics of the roll-bonding mechanism.

mode by imposing different levels of displacements by using the micrometer and measuring the load from load-cell as rollers are brought into the contact. This procedure was repeated many times with a maximum load of 1.5 kN , and a load vs deflection curve (with error-bars) is shown in the Figure 4. From these results it can be concluded that the load follows the displacement in a linear fashion with good repeatability. The stiffness of the roll-bonding mechanism based on the load vs displacement curve is estimated to be $5.3 \times 10^6 \text{ N/m}$.

ACKNOWLEDGMENT

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IMPROVEMENT OF SIMPLIFIED BIAxIAL-TENSILE-TESTING APPARATUS USING LATHE CHUCK

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INTRODUCTION

In recent years, the desired accuracy and the forming difficulty of the product size in the fabricating operation of a plate material are increasingly severe. For example, the automotive industry environmental concerns, the problem of fuel efficiency and weight reduction and so on. Reducing the weight of automobiles is one of the primary concern by which their fuel efficiency is lowered. The two basic approaches are in automotive design and in materials selection, and they are closely related. As a countermeasure, the demand of high-tensile steel or aluminums alloy sheet is increasing rapidly. Because the press forming can produce cheap parts in large quantities, it is used for the processing of the automotive material.

The biaxial tensile testing apparatus of the hydraulic control for which the estimation method of material used cruciform specimen is already developed [1]. This apparatus has measured the forming limit and yield loci of various materials with sufficient accuracy. However, in order to pursue performance, apparatus is large-sized and is complicated. Development of a simple and accurate evaluation technique is desired in small and medium enterprise or a manufacturing site.

Using the scroll chuck used for a lathe, we developed the simplified biaxial tensile test apparatus. Then, independent chuck by which each axis is independent and operates, arbitrary stress ratio condition are made to act on a specimen, and this study aims at measuring the plastic deformation characteristic with high precision of the apparatus and strain ratio after fraction of sheet metal.

NEW AND OLD TESTING APPARATUS

The old simple biaxial tensile testing apparatus which we manufactured is shown in Figure 1 [2]. This apparatus consists of three portions. They are independent chuck used for a lathe, motors (MUSCLE Cool Muscle Servo System by Muscle Co.), and four vices to the numbers 1-4 for chucking the specimen. The vices were moved by the motors controlled by the personal computer. The normal tensile speed was 5mm/min. There were some problems in this apparatus. One of them is that the rigidity of apparatus is low, because the distance of chuck and a specimen grasp device is long. The distance is 121 mm. Therefore, the moment of 121FNm is acted near a chuck root.

So, in this research, design development of new biaxial tensile apparatus was performed based on the following design concepts for the purpose of measurement accuracy improvement.

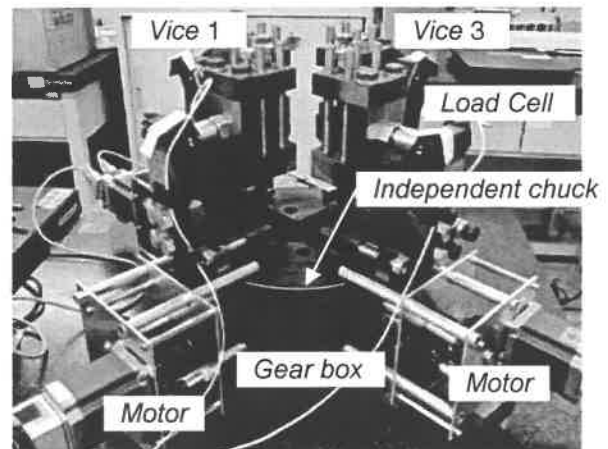


FIGURE 1 Old Simple biaxial tensile testing apparatus set on the independent chuck

- Material is used by SK105
- A maximum tensile load is 2kN
- A specimen attachment position is made lower than the height of a chuck

The plan design is shown in Figure 2, and the model designed by 3D cad is shown in Figure 3. The actually manufactured device is shown in Figure 4. Figure 5 is shown the new experimental apparatus.

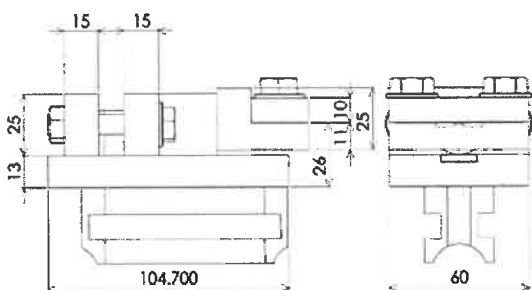


FIGURE 2 Design of New device for testing apparatus

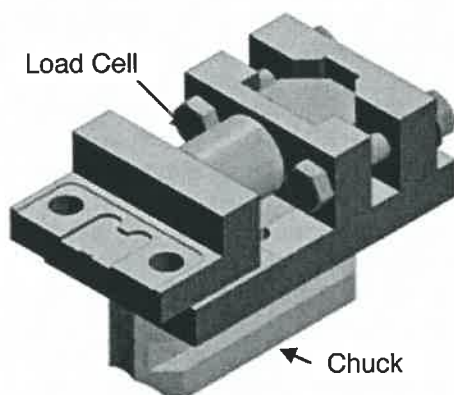


FIGURE 3 3D view of new device

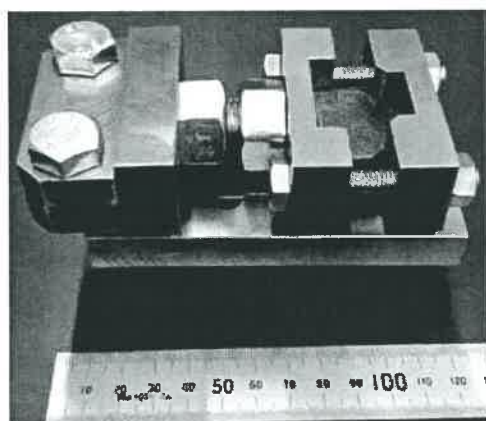


FIGURE 4 Manufactured new device

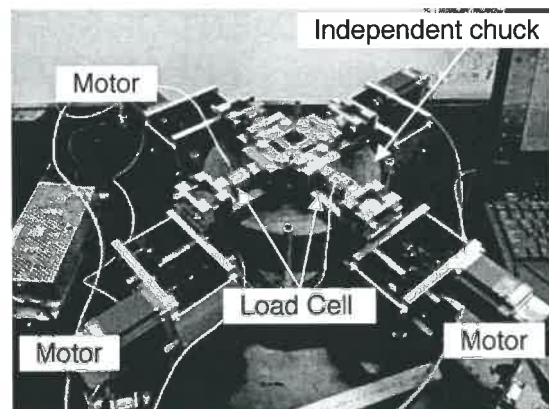


FIGURE 5 New Simple biaxial tensile testing apparatus set on the independent chuck

EVALUATION OF RIGIDITY

In order to confirm the rigidity of the old apparatus, the aluminum sheet was pulled as shown in Figure 6. Evaluation of the rigidity of apparatus is performed by calculating the displacement of a flexible region until it fractures from the start of test, and the grasp part of the old apparatus. Load was measured by the miniature load cell (ELHM-T4, MESS & SENSORTECHNIK Co.) in a figure, and was measured using the displacement from dial gage. By pulling as a result, first, while inclination was loose, it began to incline from the place exceeding 350N, the relation between load and displacement magnitude to alignment became, and it became clear that 0.48 mm of displacement has occurred in 1.4kN.

Figure 7 shows an experimental setup by new apparatus. Both measurement result of apparatus rigidity is shown in Figure 8. In the case of new equipment, displacement was measured with 2.05 μm in 1.6kN. This shows that the rigidity of equipment improved 96%, and it has been successful by us.

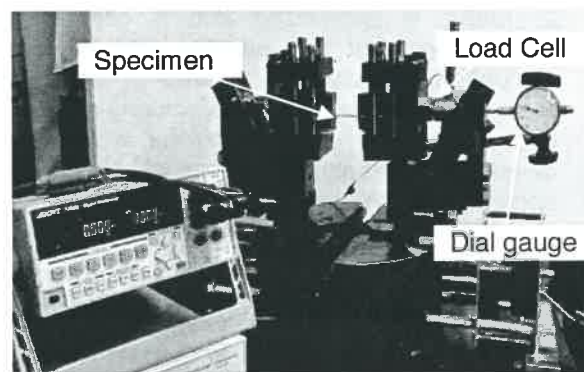


FIGURE 6 Experimental setup by old apparatus

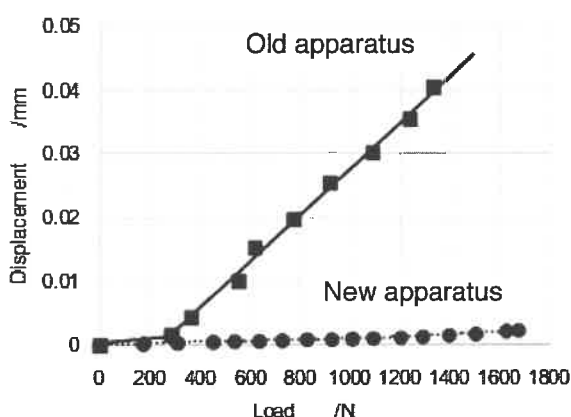


FIGURE 8 Load - displacement curve

EXPERIMENTAL PROCEDURE

Figure 9 is shown the new simple biaxial tensile testing apparatus set on the independent chuck. The vices are moved by the motor controlled by the personal computer. The normal tensile speed is 5mm/min.

The test material used in this study is an aluminum A1050-O sheet with a thickness of 1mm. The specimen of a uniaxial and a biaxial are shown in Figure 10. The true strain components, ε_x and ε_y , were measured using scribed circle as shown by Figure 11 [3].

RESULTS AND DISCUSSION

The experiment was conducted three times. The true stress - true strain curve is shown in Figure 12. It was shown that experimental equipment has the performance as well as the usual tensile testing machine.

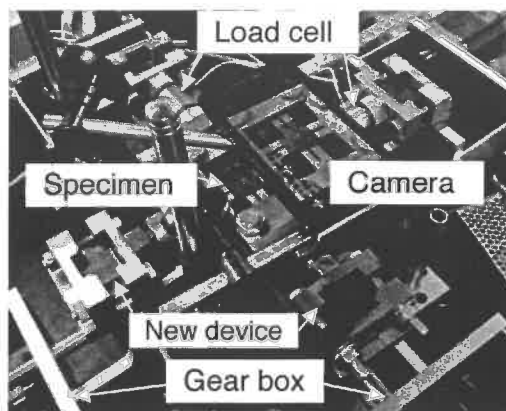


FIGURE 9 New simple biaxial tensile testing apparatus set on the independent chuck

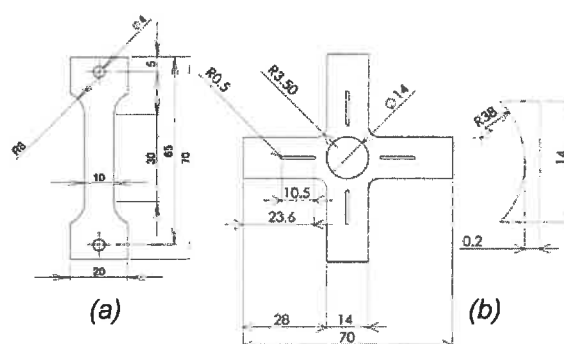


FIGURE 10 Geometry of (a) uniaxial and (b) biaxial tensile specimens

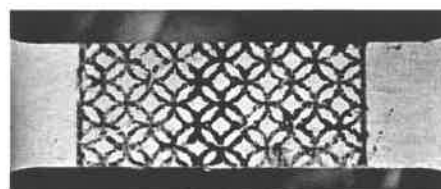


FIGURE 11 Scribed circle before deformation

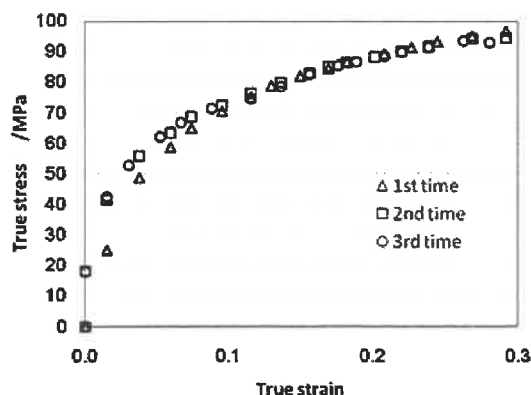


FIGURE 12 Stress-strain curve of a uniaxial tensile test

The biaxial test was carried out using the specimen of Figure 10(b). Each cruciform specimen was subjected to proportional biaxial strains with $\varepsilon_x : \varepsilon_y = 1:1, 2:1, 3:1$ and $3:2$. An experimental result are shown in Figure 13. The place where the mark is carried out with the circle is Fracture point. The forming limit diagram (FLD) showing in Figure 14 was created from each experimental result.

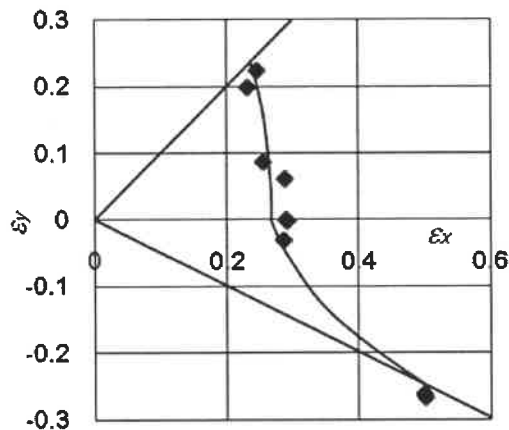


FIGURE 14 Forming limit diagram

CONCLUSIONS

In the case of new equipment, displacement was measured with $2.05 \mu\text{m}$ in 1.6KN . This shows that the rigidity of equipment improved 96%, and it has been successful by us. The uniaxial and the biaxial tensile test of A1050-O sheet metal were carried out using the simple biaxial tensile testing apparatus which operates independent lathe chuck. The stress-strain curve is obtained as good as the tensile test equipment in general. Furthermore, it was shown that it can ask also for a FLD curve with sufficient accuracy simple by having improved the rigidity of equipment.

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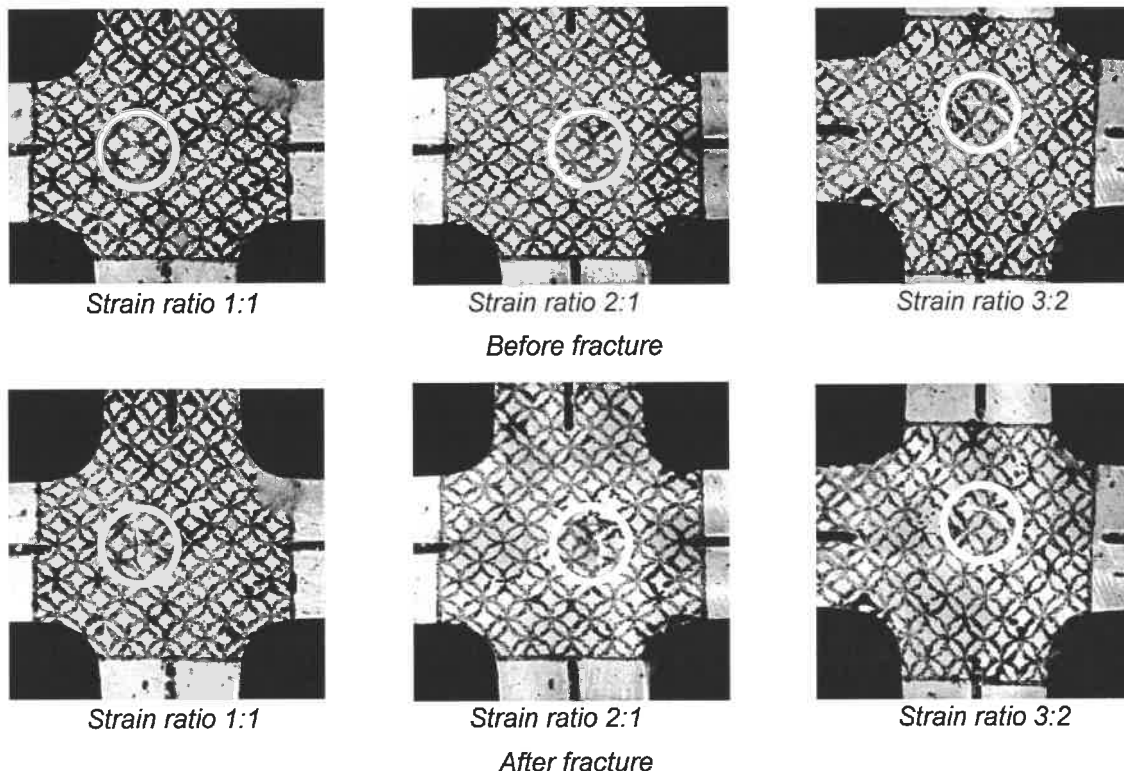


FIGURE 13 Before and after fracture of the specimens

The Analysis of Ironless Moving-Permanent-Magnet Synchronous Linear Motor Thrust Fluctuation

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INSTRUCTIONS

Ironless Moving-Permanent magnet synchronous linear motors (IMPMSLM) are widely used in precision machinery manufacturing, aerospace, microelectronics and other high positioning accuracy requirements of the occasion.

Wherein, the linear motor thrust fluctuation is an important factor affecting the positioning accuracy of the work-piece table. Many scholars have studied the thrust fluctuation reduction methods to improve the motor performance. Some articles optimized structure size [1], or the analytical method ignoring the end effect [2], or using look-up tables [3]. Another scholars have adopted active control strategy to compensate for the thrust fluctuations [4]. For ironless moving permanent-magnet synchronous linear motor(IMPMSLM), the factors that cause the thrust fluctuation include motor size, end-effect of permanent magnet array, magnetic field distribution of permanent magnet array, the number of driving phases and the form of driving current, etc. In order to achieve the purpose of reducing the thrust fluctuation, it can use compensation methods or change the motor structure size.

MOTOR STRUCTOR

In this paper, the ironless moving permanent-magnet synchronous linear motor(IMPMSLM) structure is shown in *FIGURE 1*. For the structure, the vertical coil array is as stator, and the Halbach permanent magnet array as mover.

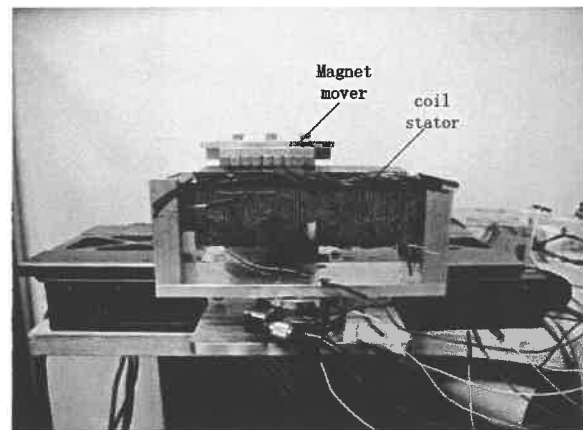


FIGURE 1 Simplify structure of ironless moving permanent-magnet linear motor

CAUSES OF TRUST FLUCTUATION

The factors that impact IMPMSLM thrust fluctuation include the size of the motor structure, the end effect of the permanent magnet array, the distribution of the magnetic field of the permanent magnet array, the number of driving phases and the form of driving current, etc. The thrust fluctuation is defined as:

$$f_w = \frac{|F_{\max} - F_{\min}|}{F_{\text{avg}}}, \text{ wherein, } F_{\max} \text{ , } F_{\min}$$

respectively means that the mover thrust maximum and minimum value, F_{avg} means that the average thrust mover.

Factor of the motor structure size

For the analysis of the IMPMSLM force model, it is essential to analyze the motor's electromagnetic field. Because the system approximates to

magnetoquasistatic field, Maxwell's equations can be simplify to

$$\nabla \cdot \vec{B} = 0 \quad (1)$$

$$\nabla \times \vec{H} = \vec{J} \quad (2)$$

For the magnetic material with the magnetization M , the conversion relationship between the magnetic flux density and the magnetic field intensity is:

$$\vec{B} = \mu_0(\vec{H} + \vec{M}) \quad (3)$$

Due to the divergence of the vector curl is identically 0, the vector magnetic potential $\vec{A}$ can be defined as

$$\vec{B} = \nabla \times \vec{A} \quad (4)$$

Recycling the expression (1) (2) (3), it can receive the Poisson equation as

$$\nabla^2 \vec{A} = -\mu_0(\vec{J} + \nabla \times \vec{M}) \quad (5)$$

Where, $\vec{J}$ is free current density.

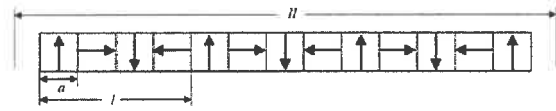
In two dimensional case, where the field lies in an x-y plane, the vector potential $\vec{A}$ is purely z-directed. In this case the vector Poisson equation simplifies to the scalar relationship

$$\left(\frac{\partial^2}{\partial x^2} + \frac{\partial^2}{\partial y^2}\right) A_z = -\mu_0\left(J_z + \frac{\partial}{\partial x} M_y - \frac{\partial}{\partial y} M_x\right) \quad (6)$$

As long as solving the magnetic vector potential $\vec{A}$, as well as using equation (4), it can obtain expression of magnetic flux density $\vec{B}$, and then get the force model of permanent magnet array [5].

The actual length of the linear motor is limited, so that the magnetic field is deformed, so there is the end-effect. The analytical model of magnetic field distribution, including the end-effect, is obtained by

using the modified Fourier series method. It sets modified pitch as ll , as shown in FIGURE 2:



l : real pitch of permanent magnet array (a mechanical period). a : the width of the vertical direction of magnetization of the permanent magnet. ll : the modified pitch of permanent magnet array.

FIGURE 2 The modified pitch of Halbach permanent magnet

According to the principle of Maxwell stress tensor, the force of the permanent magnet array can be obtained by integrating the stress tensor [5]. Since the permanent magnet array is equivalent to one period after correction, the vertical force acting on the permanent magnet array is

$$F_y = -\frac{ll \cdot w}{2} \int (B_y^2 - B_x^2) dx \quad (7)$$

The lateral force is given by

$$F_x = -\frac{ll \cdot w}{2} \int B_y B_x dx \quad (8)$$

By the analytic formula described above, in the same pitch, scanning the structural parameters of the motor including the air gap thickness, the height of the permanent magnet and the width of permanent magnet respectively, it obtains curves of thrust fluctuation, as shown in FIGURE 3, FIGURE 4, FIGURE 5 respectively:

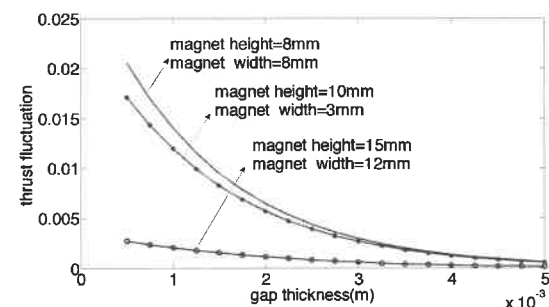


FIGURE 3 Changes of thrust fluctuation with the different thickness of gas

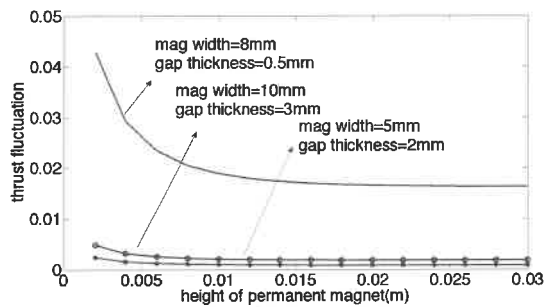


FIGURE 4 Changes of thrust fluctuation with the different height of magnet

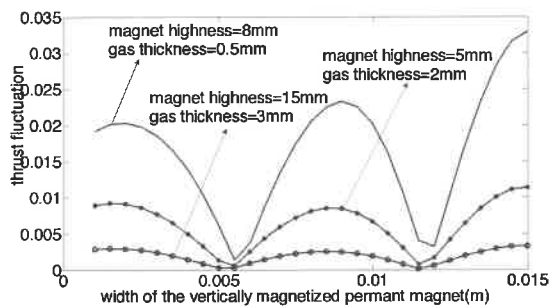


FIGURE 5 Changes of thrust fluctuation with the different width of magnet

With the increase in the thickness of the air gap, the thrust fluctuation is decreased with the index form. With the permanent magnet height increasing, the thrust fluctuation is decreased rapidly, well when the height to a certain extent, the thrust fluctuation doesn't change significantly. Along with the increase in the width of the vertical direction magnetized permanent magnet, the thrust fluctuation changes wavyly..

Factor of the distribution of the magnetic field of the permanent magnet array

As the order of the Halbach permanent magnet array is changeable, the sinusoidal form of the magnetic field generated by the permanent magnet array is corresponding changed. In this

paper, it mainly analyzes motors with two different spread patterns, as shown in **FIGURE 6**:

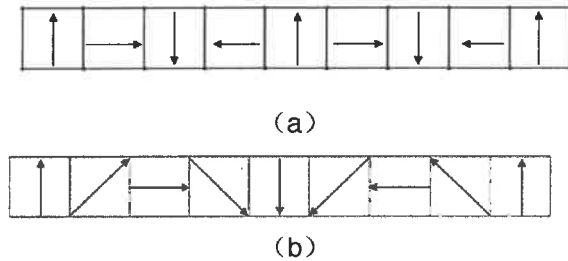


FIGURE 6 The arrangement of Halbach permanent magnet arrays

In the case of same driving current, motor forces of the two different permanent magnet arrays, are obtained by simulation as shown in **FIGURE 7**:

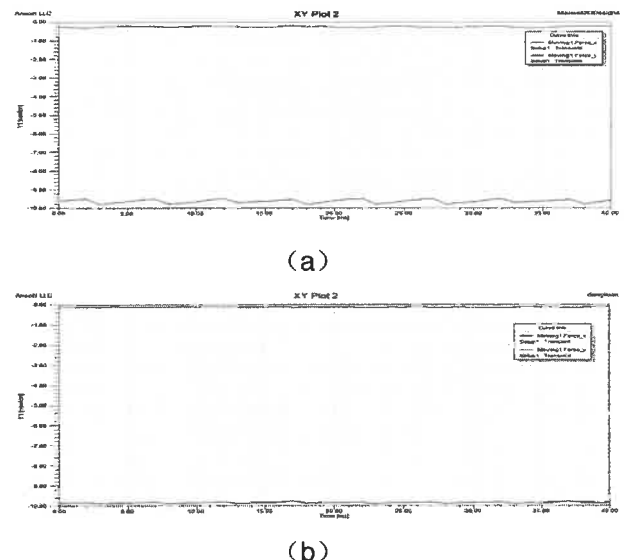


FIGURE 7 The motor force of the two arrangements of Halbach permanent magnet arrays

The thrust fluctuations, with two different arrangements of Halbach permanent magnet array, are calculated from **FIGURE 7**, respectively as: 3.43%, 1.29%. Clearly that when the magnetization direction of the permanent magnet along the 45 ° clockwise rotation distribution, the motor thrust fluctuation is much smaller. The specific reason is due to the sinusoidal form of the

magnetic field is better which is generated by the Halbach permanent magnet array with the magnetization direction of the permanent magnet along the 45 ° clockwise rotation distribution. To verify the accuracy, it analyzes the magnetic field harmonics generated by the two Halbach permanent magnet arrays, as shown in *FIGURE 8*.

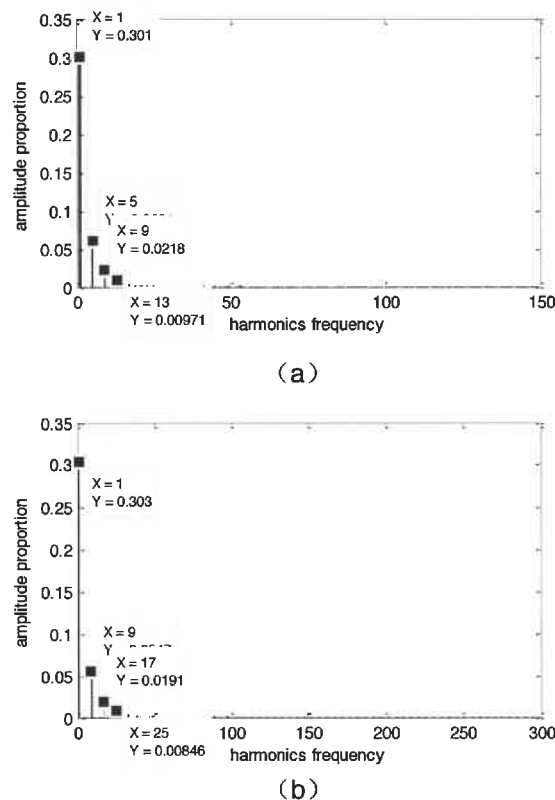


FIGURE 8 Magnetic field harmonic analysis of the two Halbach magnet arrays

Seen from *FIGURE 8*, the Halbach magnet array, with the permanent magnet magnetization direction along the 90 ° clockwise rotation distribution, produces magnetic field of fundamental, 5th, 9th,..., order, while the Halbach magnet array, with the permanent magnet magnetization direction along the 45 ° clockwise rotation distribution, produces magnetic field of fundamental, 9th, 17th,..., order, which effectively reduced the high-order harmonics of the magnetic field, thereby the thrust fluctuation is smaller.

Factor of the number of driving phases

In the case of the same permanent magnet array, the number of driving phases influences the thrust fluctuation. In this paper, it researches the thrust fluctuations when the driving currents are three-phases and six-phases.

In the case of the same Halbach magnet array, when the driving current is respectively three-phases and six-phases, motor forces are obtained by simulation as shown in *FIGURE 9*:

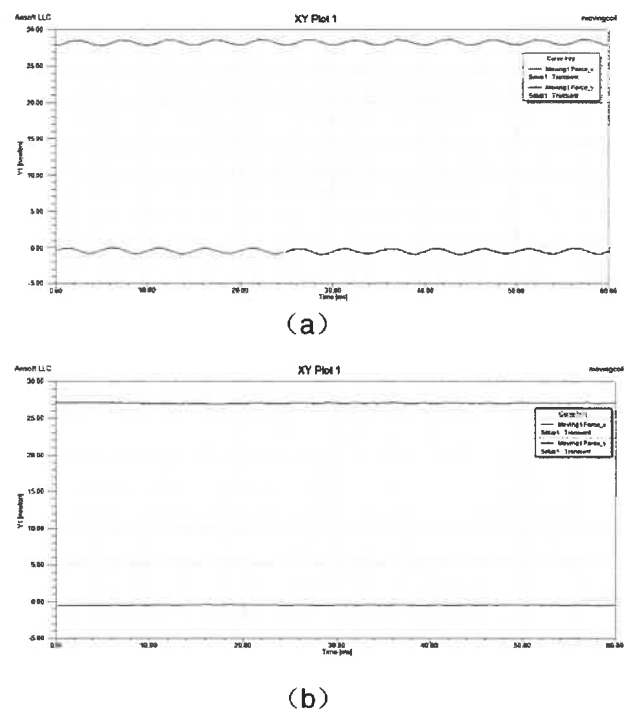


FIGURE 9 The motor force received by the 2D simulation model of the moving-permanent-magnet linear motor (a: three-phase current; b:six-phase current)

Calculating the simulation data, when through a three-phase current, the thrust fluctuation is 2.98%, well when through the six-phases current, the thrust fluctuation is 1.04%. Therefore, with the increase of the number of driving phases, the thrust fluctuation will be reduced accordingly. The

reason is that, if the coil array is equivalent to a permanent magnet array, the more the number of coils, the smaller the corresponding equivalent magnetic field harmonics, so that the thrust fluctuation decreases.

Factor of the form of driving current

For line motors, the analytic formula of the force can be expressed in different forms, for example through the motor back-EMF, then the motor force can be expressed as

$$F_y v_y = \sum_{i=1}^k E_{xi} I_i \quad (9)$$

$$F_x v_x = \sum_{i=1}^k E_{yi} I_i \quad (10)$$

Where, F_x 、 F_y is respectively the lateral force and vertical force. E_{xi} 、 E_{yi} is the back-EMF for the X and Y directions, respectively. I_i is the i th-phase current. k represents total k phases current. v_x 、 v_y is respectively means the speed of the mover in the horizontal direction and vertical direction.

There are high harmonics in back-EMFs, so through sinusoidal AC, there will be thrust fluctuation in motor. If the above expressions are as constraints, minimum power consumption that

$$\min J = \min \sum_{i=1}^k I_i^2 R \quad (11)$$

as the goal to resolve. In theory, it will get the resultant that the thrust fluctuation is 0, at the same time minimize the power consumption by the method of least squares.

When the given lateral force is 10N and vertical force is 0N, with the back-EMFs obtained by the simulation, the calculated three-phase current is shown in *FIGURE 10*. Then using this three-phases current into the two-dimensional model for simulation, it gets that 10.1N average lateral force, 0.57%thrust fluctuation, and 0.002N average vertical force, which consistent with the given force, as shown in *FIGURE 11*:

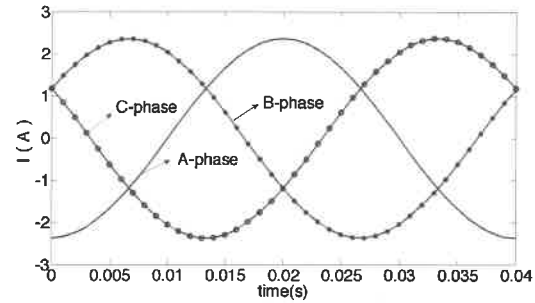


FIGURE 10 The three-phase current obtained by the back EMF

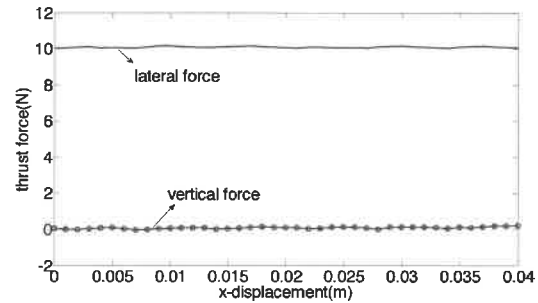


FIGURE 11 The obtained motor force when the form of current like *FIGURE 10*

CONCLUSION

The factors that cause the thrust fluctuation are analyzed in this paper, giving conclusions as follows:

- ① Considering the end-effect of permanent magnet array, the analytical model of the mover thrust force are established using the modified Fourier series. Then the effects of air gap thickness, the height of the permanent magnet, and permanent magnet width on thrust fluctuation

are analyzed. With the increase in the thickness of the air gap, the thrust fluctuation is decreased with the index form. With the permanent magnet height increasing, the thrust fluctuation is decreased rapidly, well when the height to a certain extent, the thrust fluctuations don't change significantly. Along with the increase in the width of the vertical direction magnetized permanent magnet, the thrust fluctuation changes wavyly.

②The magnetic field of permanent magnet array has high harmonics, and the less the harmonic the small the thrust fluctuation.

③In the case of the same permanent magnet array, the more phases of the coils, the smaller the thrust fluctuation.

④Due to the high harmonics of the magnetic field, the thrust fluctuation is inevitably generated when the driving current is sinusoidal. In order to solve this problem, a method of current calculation is given on the basis of the given thrust, running speed of the mover and the back-EMF, so that the thrust fluctuation can be reduced to negligible.

The analysis above can be used in practice to reduce the thrust fluctuation, and to improve the accuracy of the linear motor.

ACKNOWLEDGMENTS

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DESIGN AND TESTING OF A BACK-PRESSURE REGULATING VALVE TO REDUCE THE WATER CONTAMINATION RISK IN NEW DELHI

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PROBLEM: NEGATIVE SUPPLY PRESSURE

Although 3.7 billion people worldwide have access to piped water on their premises, there are still significant quantity and quality concerns with their water [1]. Piped water systems in poorer regions of the world often operate intermittently. The Asian Development Bank's study of 20 major cities in India found that they supplied water for an average of only 4.3 hours per day [2]. Whenever a water pipe is not positively pressured, contaminants can infiltrate through the holes and cracks in the pipe network and create significant quality concerns [3, 4]. As part of the MIT-Tata Center for Technology and Design the authors worked in collaboration with New Delhi's water utility and other private water suppliers to define and address this problem.

When such intermittently-pressurized systems are pressurized, consumers withdraw water as rapidly as possible. High flow rates increase frictional losses and decrease the system pressure. Lower system pressure means that water can no longer reach the rooftop storage tanks that many consumers use. To address this low-pressure, many consumers connect half- or one-horsepower booster pumps directly to their water supply pipe. When switched on, these booster pumps can create a suction pressure of up to eight psi.

This suction pressure creates two harmful effects. First, where the water utility's supply pressure is less than eight psi, it can induce negative pressure in the last and smallest pipe leading to the house as shown in Figure 1. Negative pressure can allow contaminants to infiltrate into this supply pipe, which water-supply engineers in New Delhi cite as the most frequent location for contamination to occur. Second, when one house uses a booster pump, the local pressure is reduced, which causes a reduction in the flow to neighbors' houses. This in turn forces neighbors to install booster pumps and further exacerbates the booster pump problem.

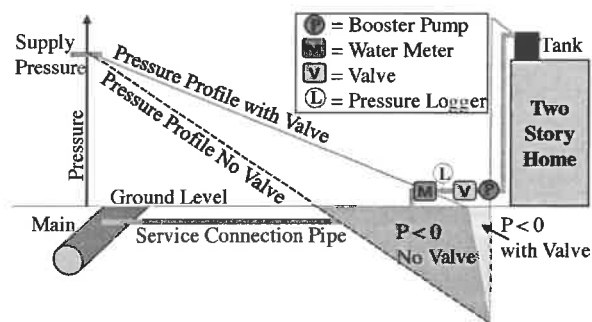


FIGURE 1. The pressure history at a typical house connection is shown both with (green) and without (red dashed) the valve installed.

SOLUTION: BACK-PRESSURE REGULATOR

One component of the negative pressure problem is booster-pump induced. This component could be mitigated with the use of a back-pressure regulating valve as in Figure 1. As the valve throttled the flow, negative pressure between the valve and the booster pump would increase, leading to cavitation in the pump. This in turn would force the pump to operate on a modified pump curve, enforcing a slower flow rate, reducing upstream frictional losses, and therefore maintaining positive upstream pressure.

Unfortunately, existing back-pressure regulators either have high frictional losses or are built for large diameter pipes and use large and expensive diaphragm actuators. In response to these limitations and with input from water system engineers, consultants, and customers in New Delhi, Bangalore, and Mumbai, the functional requirements outlined in Table 1 were determined.

KEY DESIGN FEATURE: STABILIZED STARLING RESISTOR

The key design feature of the self-actuating, full-bore, back-pressure regulating valve is a stabilized collapsing tube. The use of a collapsing tube as a regulator is frequently referred to as a "Starling Resistor" [5] after E.H. Starling, who used one in his experimental apparatus, where the tube only allowed flow through it when the upstream

TABLE 1. Summary of the functional requirements and the design parameters of the valve.

Functional requirement	Design parameter
1) Sustain positive back-pressure	Regulating valve, collapse according to: open if $P_{\text{upstream}} > 0$; close if $P_{\text{upstream}} < 0$
2) Low frictional losses when open	Full bore, minimize length, round edges
3) Reliability	Low part count, no active components, stable in throttling positions
4) Inexpensive	Low part count, common materials, no large external diaphragm
5) Withstand cyclical loading (-10psi and +30psi)	Minimize stress concentrations with gradual transitions and rounded corners
6) Burst pressure >140psi	Provide an external support for the soft tube
7) Safe for continuous drinking water contact	NSF 61(or FDA compliant for prototyping) materials
8) Stronger than standard piping	Transfer forces through an outer metal casing

pressure was greater than or equal to the pressure on its outside [6]. The outside pressure in this case is atmospheric pressure and therefore a collapsing tube would prevent negative pressure from passing upstream and would meet functional requirements 1, 2, and 4 in Table 1.

Unfortunately, such resistors are known to have an unstable regime in which they open and close rapidly [7]. Left unchecked, this behavior would induce frequent and severe pressure transients into the water system and accelerate the cyclic fatigue in the collapsing tube. No information was found in the literature about methods or mechanisms, theoretical or practical, for removing this instability. Bench-top experimentation, however, showed that the instability could be eliminated by the use of a stabilizing insert inside the collapsing tube.

Four different cross-sectional sizes were experimentally investigated (not detailed here, but in Taylor's thesis [8]). The design which minimized the instability, while still effectively regulating pressure was a cutaway pipe with a cross-section whose perimeter matched exactly the inner circumference of the collapsing tube; the cross-section is shown in Figure 2. With the collapsing dynamics stabilized, the valve met the first four functional requirement in Table 1.

DESIGN EVOLUTION

Three iterations of prototypes have attempted to render the collapsing tube concept into a field-testable prototype and are pictured in Figure 3. The alpha prototype featured a cast urethane (green) inner stabilizing layer with a half-inch in-

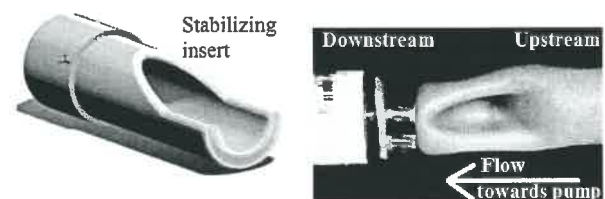


FIGURE 2. Left: Cross-section of the optimal stabilizing insert: a cutaway pipe where the cross-sectional perimeter (green) matches the inner circumference of the collapsing tube. Right: the collapsed section of the tube is concentrated towards the downstream end of the stabilizing insert and adopts the profile of the insert.

ner diameter. The collapsing tube (pink) was made of Shore A35 silicone rubber and was one-sixteenth of an inch thick. The collapsing tube was reinforced with a wire mesh tube on its outside to prevent it from ballooning at high pressure. Strain-transition layers, also of silicone rubber, of A50 were used to improve the fatigue life of the prototype. A brass crimp compressed each of the following to the stabilizing insert: the wire mesh, both transition layers, and the collapsible tube. The entire assembly was then put inside a two-inch PVC pipe assembly to discourage tampering. During field testing, 'eager' plumbers sheared the threads off five of the nine alpha prototypes. Thus, functional requirement 8 was added.

The beta prototype transferred the load through its outer metal layer made of 1.25-inch pipe. A middle reinforcing layer performed three functions: it sealed to the outer layer with two O-ring seals, it prevented the collapsing tube's balloon-

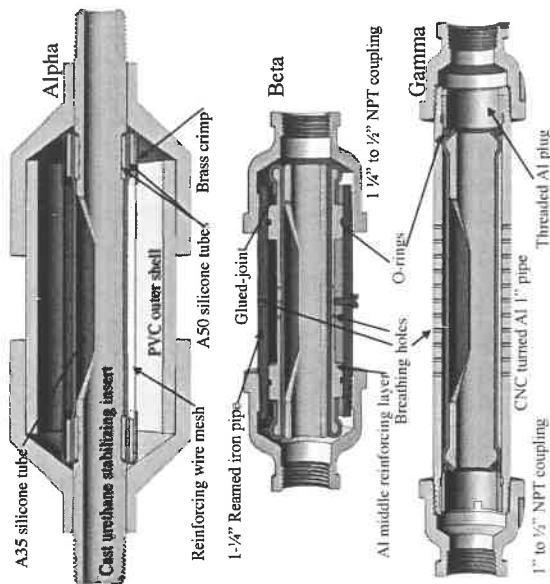


FIGURE 3. The design's evolution is shown from left to right with the alpha, beta, and gamma prototypes. Approximate scale 1:3.

ing, and it sealed to the collapsing tube through a glued joint (located on the outside of the middle layer to reduce the joint's loading). The stabilizing insert was held in place through friction. Although seven beta prototypes were successfully used in a second round of field trials in New Delhi without any instances of failure, the interface between the middle layer and the collapsing tube had a high contact area and the tube did not collapse as easily as in the case of the alpha prototype. Further, stakeholders in India from four private and public water utilities, customers in three neighborhoods, and the World Bank warned that the beta prototype was too heavy and therefore was likely to be stolen for its recycling value.

The gamma prototype addressed the additional concerns and is also pictured in Figure 3. Threaded plugs screw axially into the valve to compress the blue O-rings to form a seal between the collapsible tube and the outer metal 1-inch pipe. These seals also act to axially anchor the collapsible tube to the inner cast-urethane insert, eliminating the glued joint.

DESIGN VALIDATION: FIELD TESTS IN DELHI

Nine alpha and seven beta prototypes were tested in New Delhi, India in January and March of 2014. The most dramatic example of the valve's reduction of negative pressure is shown in Figure 4. To evaluate the performance more quantitatively, a cross-over study was conducted

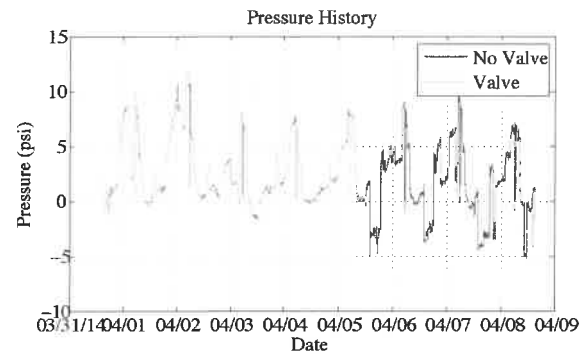


FIGURE 4. The effect of the valve at this house was to lower the duration of negative pressure by an average of 5.8 hours per day.

at 19 houses, 12 of which received 24 hours per day of water supply (unexpected by the authors). The performance was measured by the reduction in the duration of negative pressure at the connection and by a new metric for contamination risk derived in detail in Taylor's thesis [8].

The valve proved most effective at the 12 continuously-supplied connections, preventing an average of 82 minutes per day, per house of negative pressure. The valve also reduced the contamination risk at each of these connections by a median of 97% — enough to cause significant improvements in the contamination-associated health risks. The pressure regulating function of the valve was evident when pump cycles at the same houses were compared with and without the valve as in Figure 5. The valve made the most significant impact at houses with a connection pressure between three and eight psi.

The results from intermittently-supplied connections were much less clear because: four of the seven intermittently-supplied connections in the study had less than 10 minutes per day of booster-pump-induced negative pressure; the booster-pump-induced risk at the remaining three connections was still only a small fraction of the total risk; and high variability in the pressure history of these connections obscured the effects of the valve. Therefore, these seven connections had a low signal-to-noise ratio, and most conclusions about the efficacy of the valve needed to be qualified by 'while the system was pressurized' to exclude the large noise effect of the many hours per day of unpressurized pipes.

Combining the results from continuously- and intermittently-supplied connections showed that:

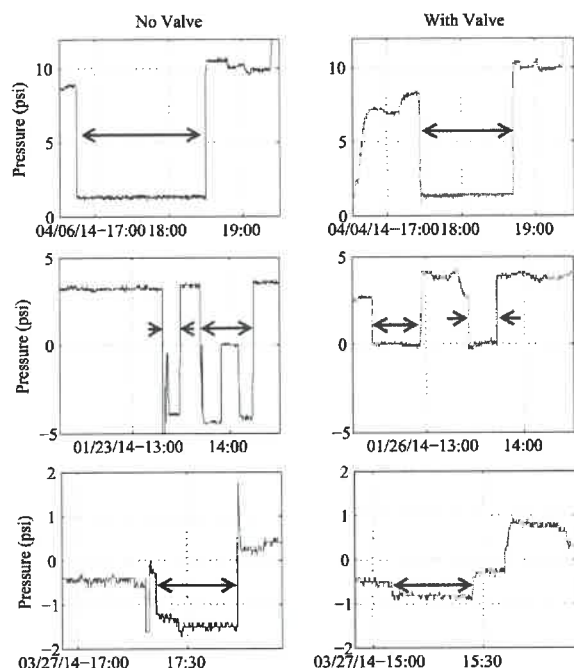


FIGURE 5. The valve's regulation effect at three houses with different supply pressures. Red arrows indicate when the pump was on. In the top two panels, there was no negative pressure to prevent; in the middle two, the valve effectively prevented negative pressure; and in the bottom two, the valve could not prevent all negative pressure because the supply pressure was negative.

- without the valve and while the system was pressurized, the contamination risk posed by booster-pump-induced negative pressure was verified as significant;
- the valve was most effective at limiting negative pressure below -1.4psi, preventing 96% of all pressure less than -1.4 psi;
- the valve reduced the duration of negative pressure from an average of 99 to 45 minutes per day, per connection while the system was pressurized;
- the valve reduced the risk of contamination by a median of 80% while the system was pressurized; and
- the valve reduced the contamination risk posed by negative pressure by two orders of magnitude at six houses — enough to correspond to significant reductions in health risks.

ACKNOWLEDGMENTS

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Development of an X-ray Interferometric and micro-CT Facility

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INTRODUCTION

X-ray imaging has been around for over a century and is largely credited as being started by William Röntgen. Since that time the most common usage for x-rays is imaging internal features of objects by comparing regions of different contrast. This method is referred to as absorption-contrast imaging and is the result of the absorption of x-ray photons. Regions of higher density absorb more photons than regions of lower density, yielding the contrast difference. There is a downside to using the absorption-contrast imaging method and that is a lack of distinguishable detail specifically in low density specimens such as biological tissue [1,2]. One solution to this problem is to measure the phase of a beam after having passed through a specimen.

In the early 1900's, a number of scientists that include Max von Laue, William H. Bragg, and William L. Bragg discovered x-ray diffraction from crystals and from this emerged the field of x-ray crystallography. Later in 1965, an x-ray interferometer constructed from single crystal silicon was demonstrated by Bonse and Hart enabling the measuring of phase [1]. The type of interferometer design used by Bonse and Hart is often referred to as a triple Laue (LLL) interferometer and since then, multiple researchers have successfully built phase-contrast imaging and computer tomography systems using various interferometer designs [3–6]. However many of these systems are built using synchrotron sources. This work is to discuss the development of a phase-contrast x-ray facility utilizing a laboratory micro-source generator.

THEORY

Phase measurement of an x-ray beam requires beam separation prior to beam combining. To do this, a single crystal material orientated about a specific plane is cut to produce three equally spaced blades known as the splitter, mirror, and analyzer (see Figure 1). When the incident beam X_i interacts with the splitter blade and

satisfies the Bragg condition at angle θ , the beam is separated in two. The diffracted beams then interact with the mirror blade and also at the diffraction angle, yielding a total of four beams between the mirror and analyzer blades. Of the four beams, two beams converge at the analyzer and produce a standing wave before exiting to a detector. Placing a specimen in the path of either the diffracted beam between the splitter and mirror or the path of a converging beam between the mirror and analyzer changes the phase of the beam of that diffracted beam.

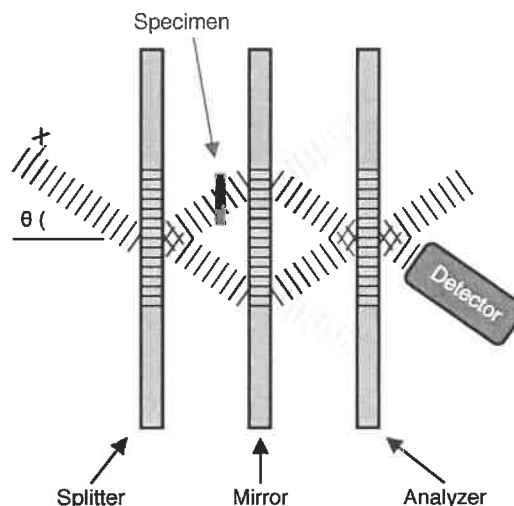


FIGURE 1. Principle of operation of the x-ray interferometer showing location of specimen for phase measurement.

The relationship between material and phase can be calculated from the equation [6]

$$\phi(x, y) = -\lambda r_0 \int_M \rho(x, y, z) dz. \quad (1)$$

Where:

λ is the wavelength of the X-ray radiation

r_0 is the classical electron radius

$\rho(x, y, z)$ is the electron number density

In the case of a homogeneous material, Equation 2 can be substituted into Equation 1.

$$\int_M \rho(x, y, z) dz = \rho_m N_A \frac{Z}{A_r} t. \quad (2)$$

Where:

t is the material thickness
 ρ_m is the material density
 N_A is Avogadro's constant
 Z is the atomic number
 A_r is the standard atomic weight

In this case, the measured phase is a function of material and specimen thickness. Hence, for a constant thickness specimen, variations in phase will correspond to material variations and vice versa.

FACILITY DEVELOPMENT

The x-ray facility discussed in this abstract is broken into five individual subsystems. The first of the subsystems is the lead-lined enclosure within which the remaining subsystems are housed for radiation safety. These remaining subsystems are described as the x-ray generator, interferometer, detectors, and specimen. At the center of the system for control and data processing is a multicore processor computer connected to an x-ray camera by camera-link, an x-ray source controller by serial communication, and a National Instruments CompactRIO (cRIO) by ethernet. The cRIO operates in FPGA mode and contains modules for analog out, analog in, digital I/O, and serial communication.

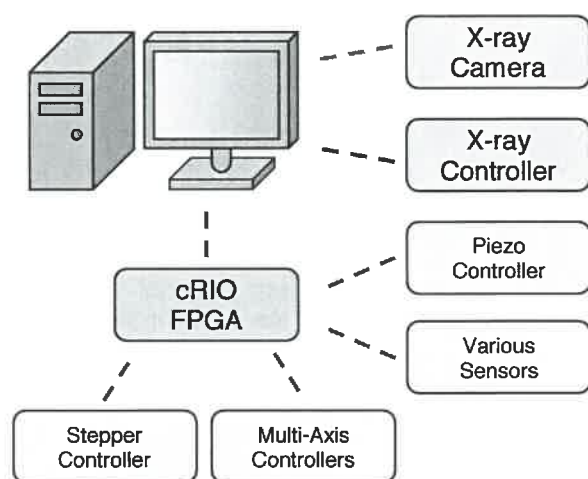


FIGURE 2. Block diagram representation of x-ray facility control hardware.

Connected to the serial module are multi-axis stage controllers and a stepper motor controller. Multiple temperature sensors are monitored by the analog in module, while the Geiger-Müller

detectors are monitored by the digital I/O. Figure 2 depicts the system configuration as a simplified block diagram.

Enclosure

To house all components, a custom lead-lined enclosure was designed and fabricated. The enclosure was designed to sit on a vibration isolation table with interlocking steel frames whereby the top and bottom frames contain the side frames. An exploded view showing the individual frames is illustrated in Figure 3. Lead-lined acrylic windows on the two front doors and sides maximize the viewing area into the enclosure and remove the need for internal lighting. Lead-lined wood is used for the ceiling and back of the enclosure. Securing the top frame to the bottom frame are four tie rods, Figure 4.

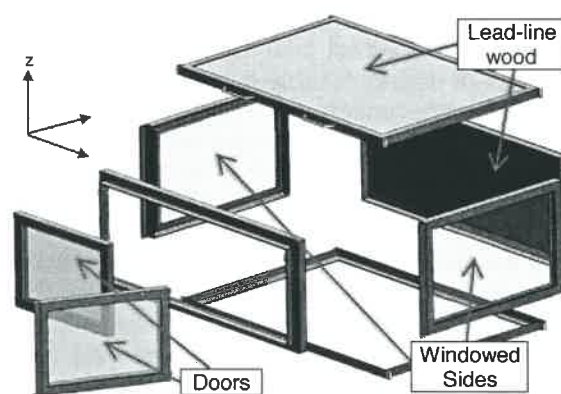


FIGURE 3. Exploded view of x-ray enclosure.

The steel frames and wood were lined with 1.5 mm thick lead sheeting and the lead identified by coating with white paint for safety. The internal dimensions of the enclosure are 1.5 m wide, 1 m deep, and 0.65 m high. A photograph of the assembled enclosure is shown in Figure 4.

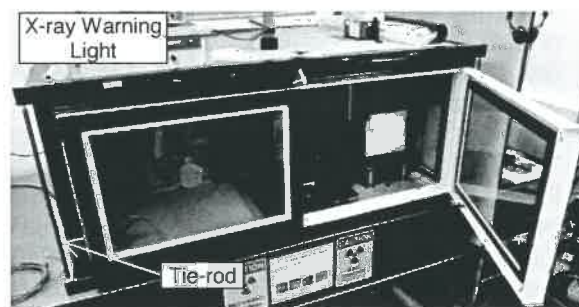


FIGURE 4. Photograph of assembled x-ray enclosure.

X-ray Generator

The x-rays are generated using a Bede microsource system with a molybdenum target. The microsource controller is set to operate up to 25 W with a voltage between 10 kV to 50 kV. An attachable optic is used to focus the x-ray to a spot size of approximately 1.8 mm FWHM at 750 mm from source. Multiple fail-safe safety interlocks exist to protect the user and equipment. For operator protection, interlock switches on the enclosure doors disable the generation of x-rays if the shutter and door are both open, while interlocks for the generator shutter lamps and external warning lamp are also present. The generator target is protected by a flow meter interlock monitoring the water cooling system to ensure flow between 350 and 600 ml·min⁻¹.

The x-ray microsource is mounted to a rotary table modified for stepper motor control. A 90:1 speed reducer connects the stepper motor to the rotary table to increase angular resolution when adjusting the generator to satisfy the diffraction condition (Figure 5).

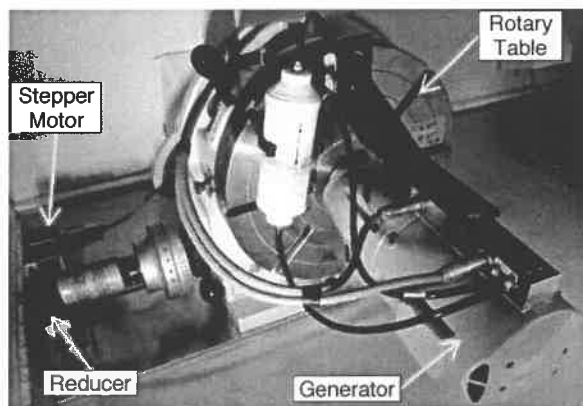


FIGURE 5. Automated goniometer for rotation control of the Bede micro-source x-ray generator.

Interferometer

Monocrystalline silicon was aligned along the (111) and (220) planes. Following alignment, three blades were cut into the silicon with a 9 mm by 8 mm cross-section. The thicknesses of the blades are 1 mm for the splitter (S) and analyzer (A), and 2 mm for the mirror (M).

After manufacturing the blades, a flexure was machined into the crystal using a high speed spindle. Surface damage produced by the machining process was removed in a HF, HCL, and acetic acid solution with approximately 50

μm removed. The interferometer with flexure is shown in Figure 6.

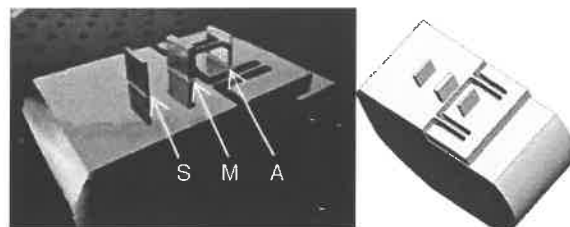


FIGURE 6. Photograph of interferometer monolith with four-bar flexure translation of the analyzer blade (left) with solid model (right).

The interferometer will sit on a kinematic mount attached to a 100 mm Z-axis linear stage (Figure 9) to position in the x-ray beam path. Control of the interferometer's flexure is currently being designed to have a linear range between 3 to 5 nm. One potential design for the flexure actuator is shown in Figure 7. A piezoelectric actuator will position a quartz tuning fork with attached fiber that acts as a sensing element until contact with the monolith is detected [7]. Once contact is established, the tuning fork would serve as an actuator for moving the analyzer flexure with attenuation of around 1:3000 being in the ratio of compliance of the analyzer flexure to that of the fiber. Flexure actuation of the interferometer in will produce a displacement of the (111) planes.

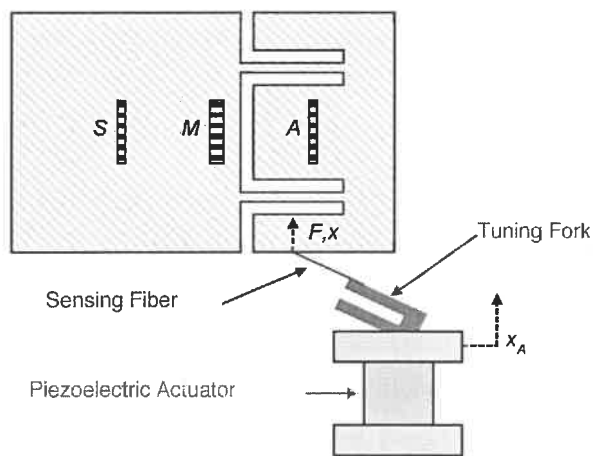


FIGURE 7. Conceptual design for a flexure actuator using a tuning fork with a sensing fiber to detect contact and apply force.

Detectors

Multiple types of detectors (Figure 8) are used in the system. The primary detector is a 1000 x 1000 pixel Photonic Science x-ray camera with 10-bit resolution capable of capturing up to 30

frames per second. The camera can be substituted for or used in conjunction with Geiger-Müller detectors capable of 25000 counts per second.



FIGURE 8. X-ray measurement using camera (left) and Geiger-Müller detector (right).

Positioning of the detectors is possible by two motorized linear axis stages, a 100 mm Y-axis and 150 mm Z-axis, and a manual rotary stage. The manual rotary stage is utilized to prevent image elongation by adjusting the camera axis to be co-linear with the x-ray.

Specimen

Positioning the specimen to be imaged is the last subsystem within the enclosure. The specimen is affixed to a precision keyless chuck with an adjustable optical post for fine positioning. These components are connected to a rotary stage that is used for creating the images used in CT reconstruction. The assembly moves along the Y-axis into the beam path by way of a 150 mm linear stage. A photograph of the assembly is shown in Figure 9.

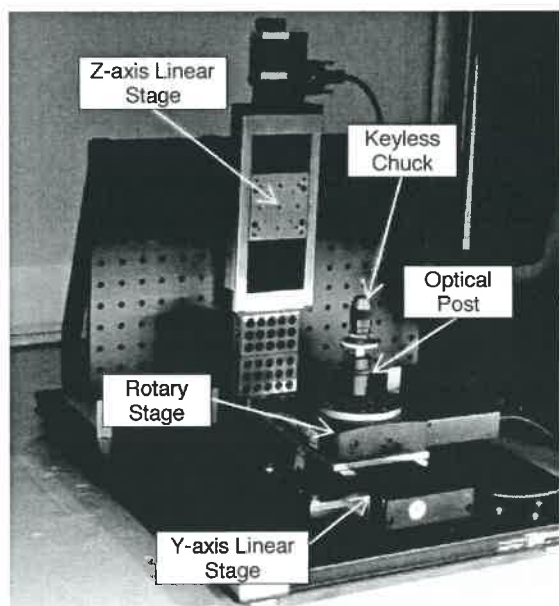


FIGURE 9. Photograph of the specimen subsystem assembly and the Z-axis linear stage used in the interferometer subsystem.

X-RAY IMAGING RESULTS

The first images taken using the facility were of a quartz tuning fork with the steel casing removed. The dimension of a quartz tuning fork is approximately 6 mm long, 1.5 mm wide, and 0.3 mm thick. Initial images showed speckling that translated into ringed patterns during cross-section CT reconstruction of multiple images of the rotating specimen. These speckles indicated the camera required calibration to correct for variations in offset and gain of individual pixels.

After calculating the offset value and producing a gain calibration map, the images were processed through a correction program. Raw and corrected images, along with reconstruction of the tuning fork cross-section from the rotated images are shown in Figure 10 and Figure 11. As seen in Figure 11, the correction reduced the speckling which translated into improved image quality.

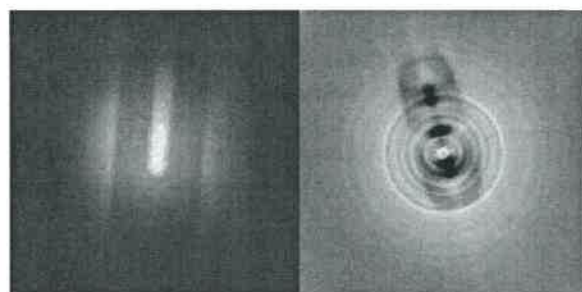


FIGURE 10. Raw image of a tuning fork (left) and the reconstructed cross-section (right).

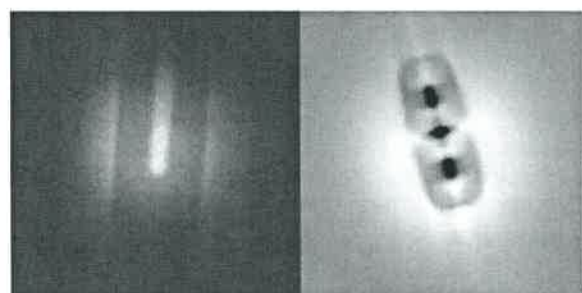


FIGURE 11. Raw image of a tuning fork (left) and the reconstructed cross-section with camera pixel gain and offset correction (right).

As this facility utilizes a focused beam, information regarding the beam's FWHM, amplitude reduction, and the cone angle were desired. Using the camera, images were taken every 5 mm over a 140 mm range. The results, Figure 12, show an amplitude decrease of $1.3 \text{ bits} \cdot \text{mm}^{-1}$ and an increase in the FWHM by $1.3 \text{ } \mu\text{m}$ per 1 mm change in distance. Using the

FWHM and displacement, the cone angle is 0.08° .

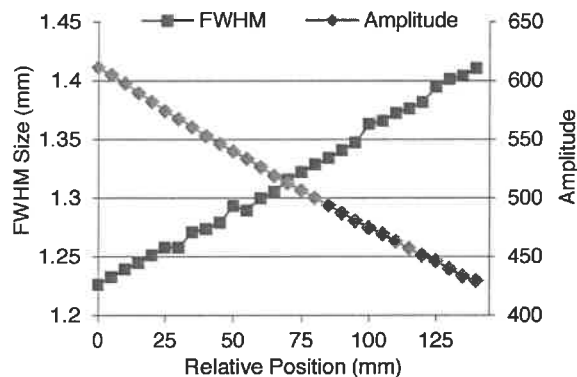


FIGURE 12. Amplitude and FWHM of the x-ray beam with respect to a relative position.

CONCLUSION

Development of an x-ray phase-contrast imaging and micro-CT facility continues to progress and evolve. The interferometer monolith was manufactured from monocrystalline silicon with a four-bar flexure and an enclosure was purposely designed and built for the project. Housed inside the enclosure are multiple linear and rotary stages and sensors for positioning the x-ray generator, interferometer, specimen, and detectors. Communicating with the controllers and monitoring various sensors is a CompactRIO FPGA, while a computer controls the x-ray source controller and camera. The facility is currently operating in absorption-contrast mode for testing and calibration purposes. These tests include imaging quartz tuning forks and reconstructing the cross-section using the inverse radon transform. Finally, measurements of the x-ray beam at various distances from the source have been evaluated to determine the beam's FWHM diameter and cone angle.

FUTURE WORK

Artifacts are being produced using deep reactive ion etching for metrological calibration of the facility [8]. These artifacts contain numerous features down to $1\ \mu\text{m}$ wide and consist of up to third-order geometric shapes. A design for securing the interferometer monolith to a vertical orientated linear stage is in progress. The design utilizes kinematic mounts to enable users to change the interferometer in the system and features an actuator for displacing the analyzer blade flexure for absolute phase reconstruction. Parallel to the facility development, a method for

automating crystal orientation is in development to enable the facility to orientate and produce interferometers using various materials and about different orientations.

ACKNOWLEDGEMENTS

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MAGNET ASSISTED STAGE FOR EFFICIENT WAFER SCANNING

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OVERVIEW

A magnet based passive-assist device for improving the motor efficiency of wafer scanners is presented. The magnetic assist device improves the motor efficiency in wafer scanning applications by providing assistive force during the acceleration/deceleration portions of the stage's motion, where the motor is less efficient in converting electricity to motor force due to copper losses. A magnetic assist device prototype was installed on an existing stage to reduce the RMS motor force by 55% and the energy consumption by 73% without negatively affecting the control precision.

INTRODUCTION

Optical lithography is the method of choice when patterning coated photoresist on silicon wafers for subsequent processes in the semiconductor industry. During the wafer scanning process, the scanning stages, which are typically driven by linear motors and supported by low friction guideways, operate at constant speeds of 0.5 to 1 m/s [1, 2]. Consequently, the actuation force requirement during scanning motion is very low. However, during motion reversals at the end of each scanning stroke, the stages have to provide high accelerations (i.e., 1 to 2 g) in order to minimize the non-scanning time and maximize throughput [2]. The linear motors therefore have to be sized to provide high peak and continuous forces, which in turn increases the size and cost of associated components (e.g., power amplifiers and support structures).

Besides its impact on machine footprint and cost, the high force requirements of wafer scanning stages have huge implications on efficiency due to copper (or resistive) losses in the motor coils [3]. Copper losses generate heat that is conducted to surrounding components causing thermal errors. Preventive measures such as forced cooling and thermal isolation must be taken to mitigate these temperature-induced errors. Copper losses are proportional to the square of the motor current; therefore it is advisable to increase the motor force to current

ratio whenever possible [3]. One way of achieving this goal is to augment the motor with a passive-assist device. The passive-assist device acts as a (dynamic) counterbalance that stores and releases mechanical energy thus reducing the power requirements from the motor. Passive-assist devices have been presented in the forms of a torsion spring attached to a revolute joint of a robot [4] and a linear spring attached to each linear axis of a planar stage [5]. In each case, the spring was carefully designed to minimize the actuation energy or peak motor current for a family of motion trajectories of the robot or stage. Even though the spring was optimal for the family of trajectories, it was shown to increase the actuation requirements for some of the trajectories within the family because the motors needed to do work against the spring. The implication is that the size of the motors would have to be increased to accommodate such trajectories. For wafer scanners, it is more advantageous to have a spring that provides assist only during motion reversals and disengages during constant velocity scanning. Moreover, given the high precision requirements of wafer scanners, it is crucial to realize passive-assist using a non-contact device. This paper presents a permanent magnet based passive-assist device (or magnetic assist device, for short) that meets the requirements for wafer scanning applications.

MAGNET BASED PASSIVE-ASSIST DEVICE

A non-contact spring, which is engaged and disengaged between the moving table and the linear motor of stage, can be realized by exploiting the repulsion force between a pair of permanent magnets of identical poles. The schematic representation in Fig. 1 shows the idea, where two pairs of permanent magnets (PM1 and PM2) are mounted at the two ends of the stage stroke and the moving table. F_M is the actuation force of the linear motor while F_{PM1} and F_{PM2} are the repulsion forces produced by PM1 and PM2, respectively. The repulsion force between two permanent magnets rapidly

increases as the gap between them reduces. Consequently, as the table approaches either end of its stroke, where motion reversals occur, the repulsion force is very high, providing the motor with the needed assist. However, as the table moves away from the stationary magnets, the repulsion force drops drastically such that it is minimal during the constant velocity scanning portion of the motion.

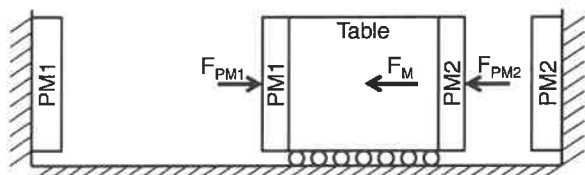


FIGURE 1. Schematic representation of magnet assisted stage where permanent magnets are mounted at ends of the stroke and moving table.

PROTOTYPE DESIGN AND EXPERIMENTS

A magnet based passive-assist device was prototyped in a 2-D Halbach arrangement to increase the force density by rotating magnet poles for the magnetic field augmentation [6]. 9.525 mm (3/8") long Neodymium magnet cubes were used for constructing 2-D Halbach arrays. As shown in Fig. 2, the device was attached to one end of an air-core linear motor driven stage (described in Ref. [7] in the context of a hybrid feed drive). The linear motor is capable of producing 1,200 N peak and 211 N continuous forces with a force constant of 57.0 N/A_{RMS} [8].

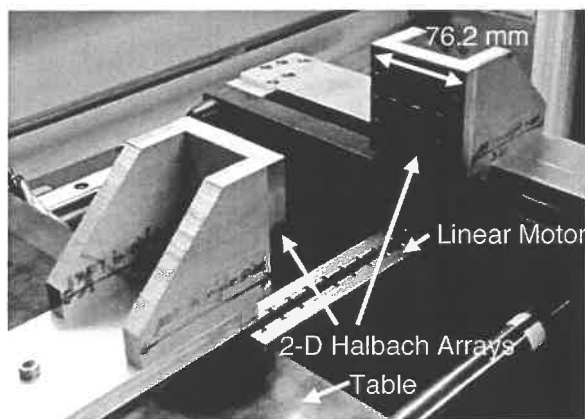


FIGURE 2. Magnet assist stage prototype where two 2-D Halbach arrays are shown to produce repulsion force for assisting linear motor during motion reversal

For measuring F_{PM} , the table was moved toward the end of stroke at a constant speed (50 mm/s) while measuring the required motor current. The motor current was converted to force to create

the relationship between F_{PM} and the gap between the two magnet arrays. The force vs. gap relationship is shown in Fig. 3, where the force is asymptotically decreasing toward 0 N with the increasing gap.

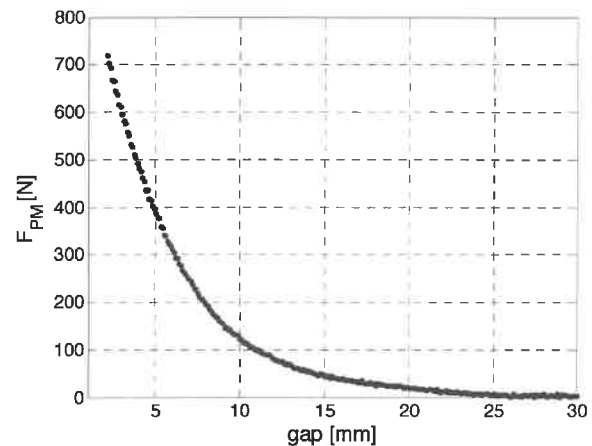


FIGURE 3. Measured repulsion force of a 2-D Halbach array pair

The trapezoidal velocity reference trajectory profile shown in Fig. 4 was used for validating the magnet assisted stage design. The trajectory had a 15 m/s² acceleration limit, 0.5 m/s scanning speed, and 97 mm stroke. This combination resulted in an 80.3 mm long constant speed scanning length.

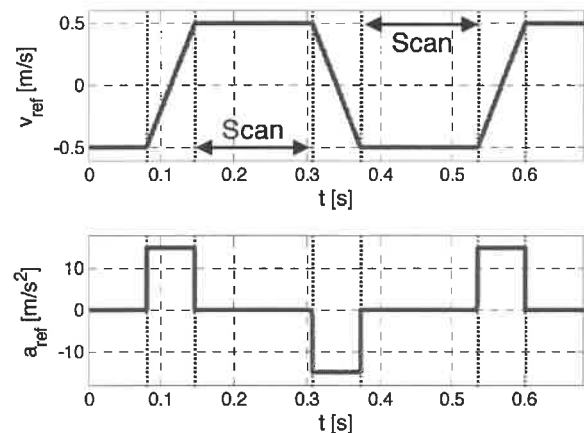


FIGURE 4. Reference trajectory used in experiments. (a) velocity profile, (b) acceleration profile. Wafer is scanned during constant speed portion of motion.

The magnet assisted stage was controlled using a traditional P-PI controller with the PI velocity (V_{FB}) and P position (P_{FB}) feedback loops combined with velocity (V_{FF}) and acceleration (A_{FF}) feed forward loops shown in Fig. 5. An

additional loop was added for feed forward compensation of the force produced by the magnetic assist device. The compensator consists of a look-up table created from the measured F_{PM} to gap relationship shown in Fig. 3. The expected distance between the two permanent magnet arrays based on the reference position command is used to provide the controller with an estimation of how much assist force is available from the magnets, so that the controller only outputs the motor force needed for the motion. In addition, the compensator helps the controller to quickly reject any spill-over disturbance forces from the permanent magnets that may otherwise adversely affect position control precision during the constant speed scanning region.

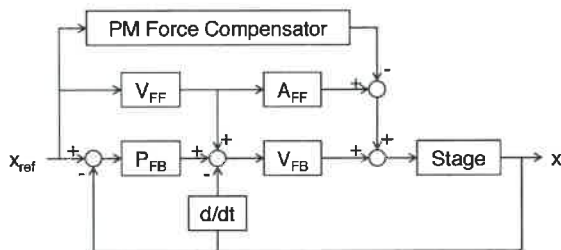


FIGURE 5. Control scheme of magnet assisted stage.

Two scanning experiments (one without and the other with the assist device) were conducted to assess the effectiveness of the assist device. The first experimental result without the assist device was used to establish the baseline stage performance. Fig. 6 shows the performance comparison between the baseline stage and the magnet assisted stage. The velocity dependent nominal guideway friction has been subtracted to show the motor force for moving the inertial load. The moving mass was estimated to be about 39.5 kg for the baseline stage and the magnet assisted stage had about 1.1 kg additional mass for extra features for mounting magnet arrays. As seen from the figure, the motor force was drastically reduced during the acceleration portion of the motion due to the assistive force provided by the permanent magnet pair. It should be noted that the motor needed to provide a small amount of compensatory force to maintain the constant speed as the moving table was approaching the stroke end due to the spill-over repulsion force from the magnets. This repulsion force is considered a disturbance from the controller perspective, but it is effectively compensated by the PM force compensator. The red dotted lines

in Fig. 7 represent the 1 μm control precision window. It can be seen that the position control precision during the scanning portion and the settling time after acceleration/deceleration did not worsen with the addition of the assist device.

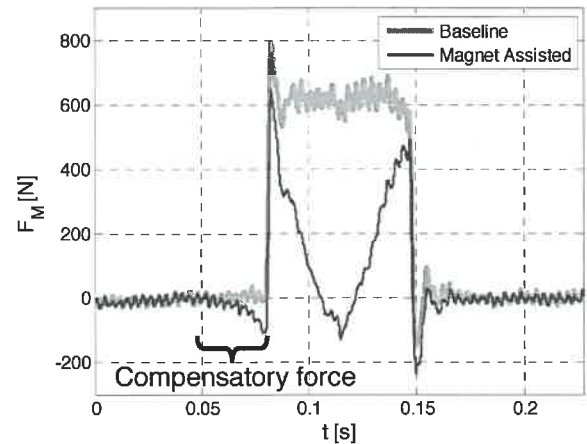


FIGURE 6. Comparison of motor force between baseline stage and magnet assisted stage during motion reversal at stroke end.

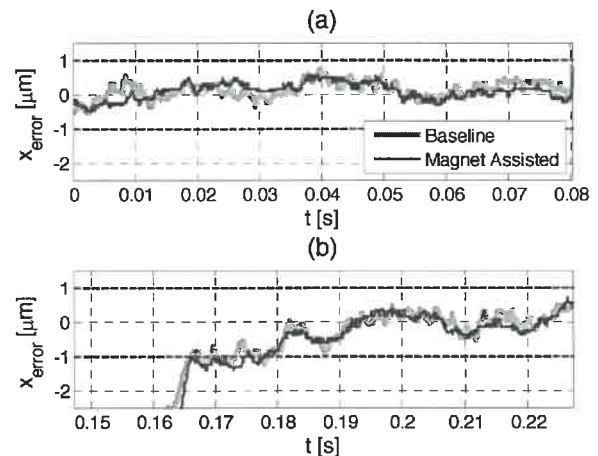


FIGURE 7. Position error of stage while (a) moving toward stroke end, (b) moving away from stroke end.

The energy (E) consumed by the motor is calculated using Eq. 1. T_a and T_b are the start time and the end time for the time window of interest, v is the velocity of the moving table, F_M is the motor force applied to the moving table, and $K_M = 21 \text{ N/W}^{0.5}$ is the motor constant associated with the copper losses [8]. Essentially, Eq. 1 considers the mechanical work for moving the inertial load and the copper loss terms in calculating the energy consumption. For the analysis of this study, $T_a = 0 \text{ ms}$ and $T_b = 227 \text{ ms}$ are used for half of the scanning motion

period shown in Fig. 4 (i.e., the half motion command near the end of the stage equipped with the assist device).

$$E = \int_{T_a}^{T_b} |v \cdot F_M| + \frac{F_M^2}{K_M^2} dt \quad (\text{Eq. 1})$$

The force and energy consumption differences between the two experimental cases are shown in Table 1. The trajectory used during the experiments demanded about 335 N_{RMS} for the baseline stage, which far exceeds the motor specification for the continuous force limit of 211 N_{RMS} [8]. This means that the baseline stage would not be able to operate at this aggressive reference trajectory since it would over-heat the motor components in the long run. A lowered acceleration magnitude or the addition of some dwell time between successive scan motions would be necessary to keep the RMS force below the continuous force limit of the motor. The magnet assisted stage required 152 N_{RMS} to follow the same trajectory and this difference in force requirement translated into the smaller energy consumed. Because the copper loss is proportional to the square of motor current, the magnet assisted stage consumed 73% less energy compared to the baseline.

TABLE 1. Comparison of motor force and energy consumption between baseline stage and magnet assisted stage

	Baseline	Magnet Assisted
$F_{M, RMS} [N]$	335	152
$E [J]$	69.4	18.7

It must be noted that the reported RMS force and energy values do not include the energy consumed to overcome the friction from the guideways because wafer scanners usually use frictionless guideways. The effect of guideways friction on the motor force and the energy consumption is minor, however. The RMS forces would have been 338 N and 158 N respectively for the baseline and passive-assist stage if the guideways friction were taken into account, implying that the energy losses due to friction were about 4.0 J and 3.7 J in each case.

CONCLUSIONS

In this paper, the idea of a magnet assisted stage design for reducing the energy lost to heat in wafer scanning applications has been presented. It has been shown, using preliminary experiments, that the addition of assistive magnet arrays to an existing stage can improve its energy efficiency by reducing motor current requirements during acceleration/deceleration. Moreover, the addition of the magnets does not negatively influence the precision of the stage during scanning. Future work will be geared towards further enhancing the efficiency, precision and flexibility of the magnet assisted stage through improvements in its mechanical design, control and motion command generation.

ACKNOWLEDGEMENT

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A NEW OPTIMIZATION METHOD TO ACHIEVE COINCIDENCE OF MODAL NODAL LINES

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INTRODUCTION

450mm wafer chuck stands for the development trend of semi-conductor manufacturing equipments. But enlarged block also brings a series of problems such as structural flexibility increase and power consumption enlargement. To meet the challenge of these problems, traditional approaches such as weight reduction and stiffness improvement are applied but the effects are limited [1].

An alternative way is to deal with the flexibility directly through independent modal space control, which induces the complexity of controller design. The more efficient way is to avoid exciting flexible modes as more as possible. This could be achieved through structural design which aims to make the nodal lines of the specified flexible modes coincide at one point where the actuator is placed [2, 3].

Rein Boshuisen [4] proposed one method to apply this idea on the wafer chuck. However, the optimization is multi-objective, which causes the optimization process rather time-consuming and even unstable. In this paper, we proposed a new single object optimization method on the design of wafer chuck. This method is simpler and more effective. Calculation and experiment is provided to prove the validity of the proposed design.

MODAL APPROACH

For a linear time-invariant system, state variables such as displacement and velocity in physical coordinates could be transferred to uncoupled modal coordinates, design and control the system under modal coordinates is called modal approach [5]

Transfer Function

The dynamics of a free wafer chuck under modal coordinates can be expressed by the transfer

function of a as Eq. (1). Where $Z(s)$ is z-direction displacement of the wafer chuck, $F(s)$ is the exciting force at the location of actuator[6].

$$\frac{Z(s)}{F(s)} = \frac{\varphi_1(z_s)\varphi_1(z_a)}{m_1s^2} + \sum_{n=2}^{\infty} \frac{\varphi_n(z_s)\varphi_n(z_a)}{m_ns^2 + c_ns + k_n} \quad (1)$$

m_n : modal mass, subscript 1 indicates rigid body Mode

$\varphi_n(z_s)$: sensing ability

$\varphi_n(z_a)$: actuating ability

c_n : modal damping

k_n : modal stiffness

The product of $\varphi_n(z_s)$ and $\varphi_n(z_a)$ determines the excitability of the associated mode, $\varphi_n(z_s)$ and $\varphi_n(z_a)$ are related to modal eigenvectors. When the actuator is placed close to the n^{th} node line, $\varphi_n(z_a)$ approaches to zero, the n^{th} mode excitability will be dramatically reduced.

Position of Actuators

As shown in Fig. 1, the wafer chuck to be designed is simplified as a rectangular plate. In case of the 1st flexible mode, the nodes compose two cross dash lines. When actuators are positioned at dash line, the actuator contribution of the 1st mode will be zero. Therefore the 1st mode will not be visible in the transfer function. Another way to avoid exciting a mode is when more actuators are used with opposite modal contributions. The total contribution of that mode is cancelled out so the mode is not excited. This actuation principle is called symmetric actuation.

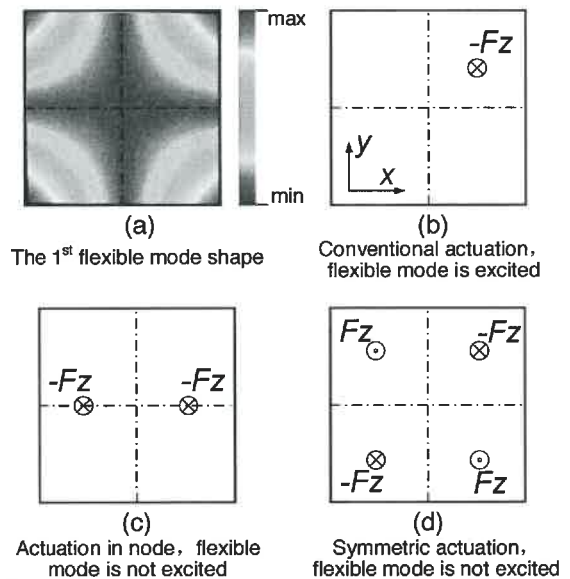


FIGURE 1. Effect of actuator location on excitation of a flexible mode

OPTIMIZATION

We designed a scaled wafer chuck which is $200 \times 200 \times 10$ mm³ to reduce the experimental cost.

Modal Nodal Lines of Original Chuck

As illustrated in Fig. 2(a) the nodal lines of the first five flexible modes do not coincide.

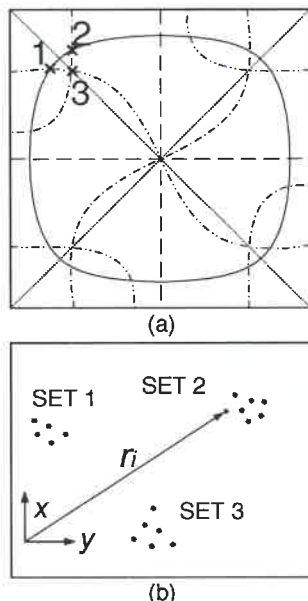


FIGURE 2. The first five modal nodal lines of wafer chuck (a) and enlarged drawing of common nodes of the 3rd & 4th, 3rd & 5th and 2nd & 5th nodal lines (b).

Optimization Fuction

In order to do the coincidence, the distribution of the nodal lines of the first five flexible modes has to be changed. For this purpose, a cavity is inserted in the middle of the chuck, the size (diameter D and depth h) could be optimized to realize the coincidence of nodal lines (Fig. 3).

Mathmetical Model of Optimization

One or more optimization objective functions (OOFs) with variables D , h must be defined to describe the coincidence degree of modal nodal lines mathematically. So when the value of OOF(s) is minimized, it directly means modal nodal lines coincide. The optimization procedure can be illustrated as:

$$\begin{aligned} &\min F(D, h) \\ &\text{s.t.} \begin{cases} 200 - D > 0 \\ D \geq 0 \\ 10 - h > 0 \\ h \geq 0 \end{cases} \end{aligned} \quad (2)$$

where $F(D, h)$ denotes the set of the optimization objective functions (OOFs).

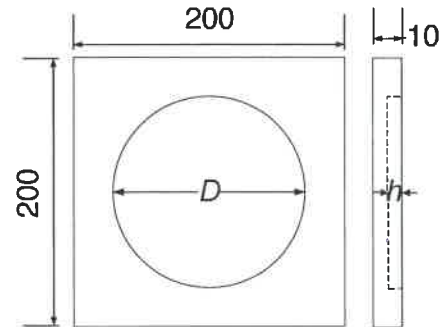


FIGURE 3. Optimization design of simplified wafer chuck

Optimization Objective Function

The quality of the optimization result depends on the choice of OOF(s). Rein Boshuisen [4] provides an OOF, which defines the sum of the norms of 3-direction (z , R_x , R_y) generalized input vectors in the modal coordinate as optimization object. This object actually is the weighted sum of 3 object functions those must be minimized. The calculation of multi-object optimization is far more complicated than single object optimization and the result is less stable.

To define a simpler and more stable single objective OOF, we introduce the variance of location vectors of common nodes of the 3rd & 4th, 3rd & 5th and 2nd & 5th nodal lines in Cartesian

coordinate as the objective function. (see Fig. 2(a) as 3 black Xs), which is expressed as:

$$\sigma^2 = \frac{1}{3} \sum_{i=1}^3 (r_i - \bar{r})^2 = \frac{1}{3} \sum_{i=1}^3 [(x_i - \bar{x})^2 + (y_i - \bar{y})^2] \quad (3)$$

where $\bar{x} = \frac{1}{3} \sum_{i=1}^3 x_i$ and $\bar{y} = \frac{1}{3} \sum_{i=1}^3 y_i$, x_i and y_i are obtained by FEM calculation.

Modified OOF for Calculation

Since FEM is based on finite discrete theory, it is impossible to extract a continuous exact node line. To solve this problem, we define the nodes whose eigenvectors values are below 0.3 as modal nodes. Then the FEM calculation result of the first five modal nodal lines of wafer chuck are slightly wider than these lines shown in Fig. 2(a), thus each of the 3 black Xs in the upper left corner of the Fig. 2(a) does not stand for one node but a set of nodes. Enlarged drawing of the region 3 black Xs are located, shown as Fig. 2(b). And Eq. (3) becomes:

$$\begin{aligned} \sigma^2 &= \frac{1}{3} \sum_{i=1}^3 \sum_{j=1}^{n_i} (r_j - \bar{r})^2 \\ &= \frac{1}{N} \sum_{i=1}^N [(x_i - \frac{\sum x_i}{N})^2 + (y_i - \frac{\sum y_i}{N})^2] \end{aligned} \quad (4)$$

where n_i is the number of nodes in SET i , $N = n_1 + n_2 + n_3$.

RESULT

With σ^2 getting smaller, the coincidence degree of the 2nd to 5th modal nodal lines apparently is getting higher. The optimization result shows that the smallest optimization objective function σ^2 is achieved when D equals 191.2mm and h equals 3.5mm. The nodal lines of the first five modes after optimization extracted from FEM are illustrated in Fig. 4.

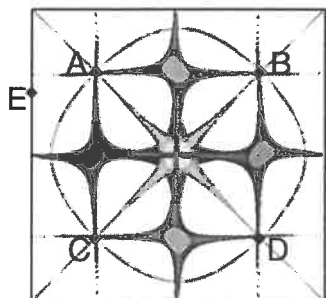


FIGURE 4. 2nd ~5th nodal lines of optimized chuck (FEM calculation result)

The 2nd ~5th nodal lines coincide at A, B, C, D. Place actuators on these 4 points, we can avoid the excitation of the 2nd~5th modes and eliminate the 1st mode through the effect of over-actuation.

PERFORMANCE

To illustrate the effect of the modal nodal lines optimization and over-actuation, the acceleration frequency response tests and impulse response tests of wafer chucks, the original and optimized, are conducted.

Acceleration Frequency Response

To improve the accuracy of the tests, the exciting and measuring locations were exchanged. The sensors were located at point A, B, C, D as shown in Fig. 4—in the test of original chuck, only one sensor was located at point A. And the excitement was located in point E which is not the node of any first five flexible modes.

Because of the symmetry of the frequency response function matrix, the test result should not be changed. Original and optimized chucks are shown in Fig. 5 (a)&(b), while comparative frequency response tests are shown in Fig. 5 (c)&(d).

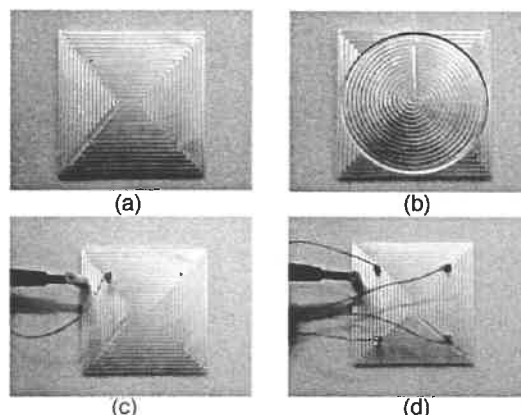


FIGURE 5. Acceleration frequency response tests of original and optimized wafer chucks

The acceleration frequency response curves are compared as shown in Fig. 6. We can see that except the first resonance, the 2th~5th resonances have been greatly reduced due to nodal lines optimization.

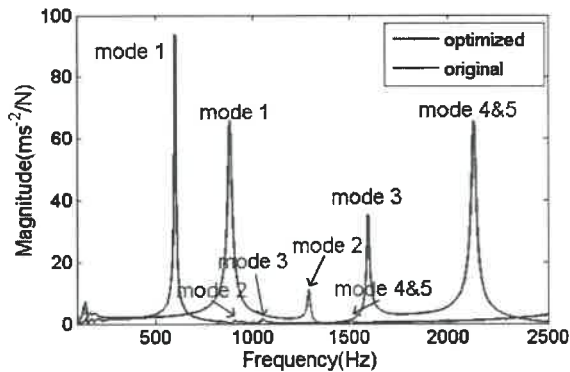


FIGURE 6. Acceleration frequency response of the original and optimized wafer chucks

Impulse Response

The sensors were located at point A as shown in Fig. 4. And the excitement was located in point E as same as the frequency response tests. With the same excitement, the impulse response of the original and optimized chucks is shown in Fig. 7.

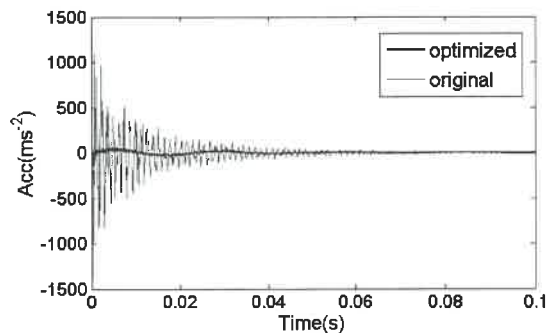


FIGURE 7. Impulse response of the original and optimized wafer chucks

The peak of optimized chuck's acceleration impulse response is just 50 ms^{-2} while the peak of original chuck reached 1100 ms^{-2} . This proved that the design has greatly reduced the flexibility of the chuck.

CONCLUSION

The critical problem in the development of wafer stage is to deal with the flexible deformation since it causes vibration and control instability. To reduce the actuating ability of the 2nd~5th modes, structural optimization is executed so that the

nodal lines of the four modes coincide; as for the first mode, over-actuation is adopted so that this mode does not deteriorate the dynamic behavior. In this paper, we have introduced the conception of variance of location vectors of common and used it to describe the dispersion degree of the 2nd to 5th modal nodal lines the optimized topology of wafer chuck has been obtained by using FEM. The frequency response tests and impulse response tests have demonstrated that the wafer chuck after design behaved more like a rigid body, which indicated a great improvement in dynamic performance. The optimization method has been proved to be correct, and could be used in optimization of 450mm or larger wafer chucks.

THANKS

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DESIGN AND ANALYSIS OF IRONLESS PERMANENT MAGNET LINEAR SYNCHRONOUS MOTOR WITH LITTLE FORCE RIPPLES FOR ULTRA-PRECISION POSITIONING SYSTEMS

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INTRODUCTION

Ironless permanent magnet synchronous linear motor (ILPMLSM) are widely applied in the field of ultra-precision positioning systems [1]. The accuracy of the model of the ILPMLSM affects the performance for positioning. Therefore, the high accurate model of the force is necessary.

In the analysis of a Halbach magnet array, an analytical model is presented by using of the magnetic scalar potential. The force between coils and magnets can be calculated with the Lorentz force law [2]. In Kim's analysis, surface-mounted linear motors consisting of permanent magnets and ironless current-carrying coils are treated in a uniform way via the magnetic vector potential. The force between coils and magnets can be calculated with the Maxwell stress tensor [3]. This method does not depend on the structure of motor. And when the force is calculated, tedious volume integration can be translated into surface integration. So far, these two methods are applied in many researches [4-10].

The models of the force in these methods are the sum of the fundamental and the higher harmonics in force. As the model of ILPMLSM is evaluated every servo time in the real-time control, a simple analytical model is used, which only consists of the fundamental and omits the higher harmonics. Therefore, one main reason for the force ripples is higher harmonics [4], [5].

The higher harmonics of force is caused by the distribution of current in windings and the pattern of magnet array. The two components have studied by some researchers, respectively. The effect of magnetizing patterns of the permanent magnets has been investigated in detail by using FEM and the simulation-based DOE method [5]. An effective ripples reduction can be achieved

by varying the parameters of the NS magnet array, such as magnet height, magnet width, air gap length and motor width [6], [7]. Force ripples for different magnet width/pole pitch in Halbach magnet array are studied by 2-D Finite Element analysis [1], [8]. A stair-step-shaped magnetic pole structure is proposed to shape the air gap flux density distribution to be as close to a sine wave as possible for the reduction of force ripples [9]. The overlapping windings and trapezoid windings can produce single-sided and perfectly sinusoidal magnetic field compared to the film windings [8], [10].

In these researches about the force ripples in ILPMLSM, one special structure of windings or magnet array is studied individually. The result is difficult to be applied in the analysis and design of other structures. And the influences of windings and magnet array are studied separately in the researches. The relationship between them is not considered. Moreover in Kim's analysis, the fundamental of the force is studied but the higher harmonics are not.

In this paper, the expression of the higher harmonics of force is derived via Maxwell stress tensor, firstly. Further, a novel design theory of ILPMLSM is proposed, which easily meets the requirement of little force ripples by eliminating higher harmonics as more as possible. Then the force ripples in a type of ILPMLSM with particular structure is analyzed. The influences of the distribution of current in windings and the pattern of magnet array are both researched. And the relationship between them is also considered. The results are validated by comparison of the ILPMLSM with finite element analysis (FEA).

ELECTROMAGNETIC MODELING

The geometry used to model the ILPMLSM is shown in Fig.1. The lower region of thickness Γ represents the winding with y-directed current density $\mathbf{J}$, which is expressed as an infinite Fourier series through terms J_{yn} . The upper region of thickness Δ represents the magnet array with magnetization $\mathbf{M}$ in the xz-plane, which is expressed as an infinite Fourier series in z-directed and x-directed through terms M_{zn} and M_{xn} , respectively. The base coordinate frame is carried by the winding. And a primed coordinate frame is carried by the magnet array, which is displaced from the base coordinate frame by a vector $(x_0 + \Gamma)\mathbf{i}_x + z_0\mathbf{i}_y$. Thus x_0 is the air gap, and z_0 is the lateral displacement of the magnet array relative to the winding. The pitch of the motor is l , and the spatial wavenumber of the n th harmonic is $k_n = 2\pi n/l$. We further define $\gamma_n = |k_n|$. The motor is assumed to be of depth w in the y-direction and of length L in the z-direction. End effects in this model are omitted.

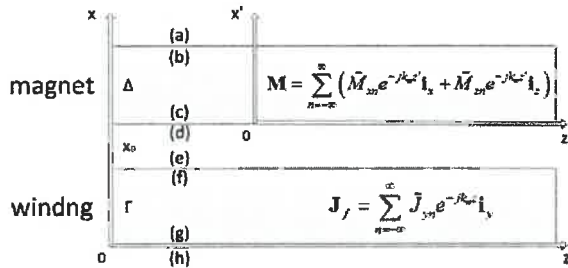


FIGURE 1. ILPMLSM model.

In this model, the expression of the higher harmonics of force is derived with magnet vector potential and Maxwell stress tensor. The force acting on one pitch of the magnet array as

$$\begin{bmatrix} F_{zn} \\ F_{xn} \end{bmatrix} = G \begin{bmatrix} \sin \gamma_n z_0 & -\cos \gamma_n z_0 \\ \cos \gamma_n z_0 & \sin \gamma_n z_0 \end{bmatrix} \begin{bmatrix} \tilde{J}_{ynr} & -\tilde{J}_{yni} \\ \tilde{J}_{yni} & \tilde{J}_{ynr} \end{bmatrix} \begin{bmatrix} \tilde{M}_{xnr} - \tilde{M}_{xni} \\ \tilde{M}_{znr} + \tilde{M}_{zni} \end{bmatrix} \quad (1)$$

where F_{xn} and F_{zn} are the n th order higher harmonic of the x-directed and z-directed forces per spatial wavelength, respectively. Terms J_{ynr} and J_{yni} represents the real part and the imaginary part of term J_{yn} , respectively. Similarly, terms M_{znr} and M_{zni} represents the real part and the imaginary part of term M_{zn} , and terms M_{xnr} and M_{xni} represents the real part and the imaginary part of term M_{xn} . The constant

$$G = -\frac{\mu S}{\gamma_n} (1 - e^{-\gamma_n \Delta}) (1 - e^{-\gamma_n \Gamma}) e^{-\gamma_n z_0}$$

contains the effects of the motor geometry. If $n=1$, the expression represents the fundamental of force.

The analysis of the expression (1) shows that the amplitude of the n th order higher harmonics of force is determined by the n th order Fourier coefficient of current density and magnetization, and independent of others. The n th order higher harmonics of force is eliminated when the n th order Fourier coefficient of current density or magnetization is zero. Therefore, a novel design theory of the motor with little force ripples is presented. Firstly, increasing the number of the zero Fourier coefficient of current density and magnetization is necessary. Then, the orders of the zero Fourier coefficient of current density and magnetization is different.

There're force ripples in ILPMLSM because the distribution of current density and magnetization are non-sinusoidal. If the number of zero Fourier coefficients of current density and magnetization are more, the distribution of current density and magnetization are close to sinusoidal. And the force ripples are reduced.

ANALYSIS OF THRUST RIPPLES OF ILPMLSM

Structure of ILPMLSM

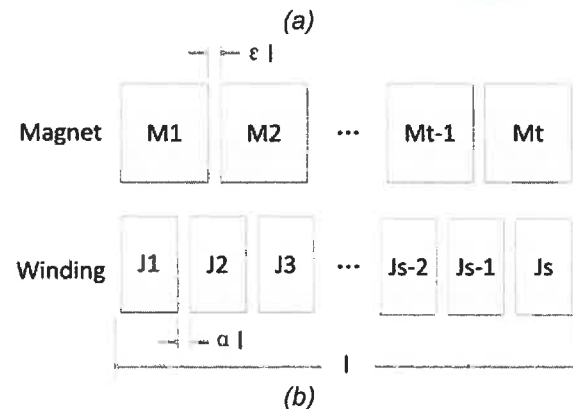
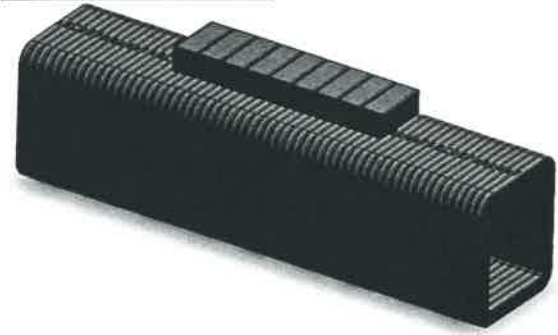


FIGURE 2. The structure of ILPMLSM. (a) Schematic view of ILPMLSM. (b) Analysis model of ILPMLSM.

In this paper, the ILPMLSM which structure shown in figure 2 is researched. Windings wrap around the stator core and are stacked side by side in this type of motor. The windings and the magnet array are divided into s th and t th per spatial wavelength, respectively. The clearance between neighboring windings is α times of pitch, and the phase angle of current in neighboring windings differs $2\pi/s$. The clearance between neighboring magnets is ε times of pitch, and the angle between the directions of magnetization in neighboring magnets is $2\pi/t$. The amplitude of the current density in winding is J_0 . And the magnetization of the magnet is M_0 .

Force Ripples

As the phase angle of current in the first winding on the left side of the selected period is ϕ , the function of the current density varied with z_0 in each winding from left to right in one spatial wavelength is

$$J_{iy} = J_0 \sin \left(k_1 z_0 + \phi - \frac{i-1}{s} 2\pi \right)$$

The angle between the vertical direction and the direction of magnetization in the first magnet on the left side of the selected period is θ , the magnetizations in z -direction and x -direction of each magnet from left to right in one spatial wavelength is

$$\begin{aligned} M_{iz} &= M_0 \sin \left(\theta + \frac{i-1}{t} 2\pi \right) \\ M_{ix} &= M_0 \cos \left(\theta + \frac{i-1}{t} 2\pi \right) \end{aligned}$$

The n th order higher harmonics of force which is produced by this type of motor derived by expression (1) is

$$\begin{aligned} F_{xn} &= \begin{cases} -D \cos \Phi, n = ps+1 \& qt+1 \\ D \cos \Theta, n = ps-1 \& qt+1 \\ 0, \text{else} \end{cases} \\ F_{zn} &= \begin{cases} -D \sin \Phi, n = ps+1 \& qt+1 \\ D \sin \Theta, n = ps-1 \& qt+1 \\ 0, \text{else} \end{cases} \end{aligned}$$

where F_{xn} and F_{zn} are the n th order higher harmonic of the x -directed and z -directed forces per spatial wavelength, respectively. The constant

$$\begin{aligned} \Phi &= \frac{2(n-1)z_0}{l} \pi + \frac{n-1}{s} \pi - \phi - \frac{1}{s} \pi - \theta + \frac{1}{t} \pi \\ \Theta &= \frac{2(n+1)z_0}{l} \pi + \frac{n+1}{s} \pi + \phi + \frac{1}{s} \pi - \theta + \frac{1}{t} \pi \\ D &= A \cdot f \cdot g \cdot h \end{aligned}$$

where

$$\begin{aligned} A(w, L, l) &= \frac{\mu w L s t J_0 M_0}{4\pi^3 \pi^3} \\ f(s, \alpha, n) &= \sin \left(\frac{n}{s} \pi - \alpha n \pi \right) \end{aligned} \quad (2)$$

$$g(t, \varepsilon, n) = \sin \left(\frac{1}{t} \pi - \varepsilon n \pi \right) \quad (3)$$

$$h(n, \Delta, \Gamma, x_0) = (1 - e^{-\gamma_s \Delta}) (1 - e^{-\gamma_s \Gamma}) e^{-\gamma_s x_0}$$

contains the effects of the motor geometry. If $n=1$, the expression represents the fundamental of force.

Analysis of the Higher Harmonics of Force

In this paper, we focus on z -directed force. The analysis method and result of the x -directed force are similar to z -directed force. Firstly, the higher harmonics is analyzed in a simple situation that there's no clearance between neighboring windings or in magnet array. That is $\alpha = 0$ and $\varepsilon = 0$. The relationship between the amplitude of the $(n+1)$ th order harmonics and the n th order harmonics in force is

$$\frac{\tilde{F}_{zn(n+1)}}{\tilde{F}_{zn}} = \frac{n^3}{(n+1)^3} e^{-\frac{2\pi}{l} x_0} \frac{1 - e^{-\frac{2(n+1)\pi}{l} \Delta}}{1 - e^{-\frac{2n\pi}{l} \Delta}} \frac{1 - e^{-\frac{2(n+1)\pi}{l} \Gamma}}{1 - e^{-\frac{2n\pi}{l} \Gamma}}$$

where $F_{zn} \neq 0$. It can be proved that the ratio above is less than 1. In another word, the higher the order of non-zero higher harmonic is, the smaller its amplitude is. Therefore, the first order non-zero higher harmonic is the main component of the force ripples caused by higher harmonics.

The relationship between the amplitude of the n th order harmonics and the fundamental in force is

$$\frac{\tilde{F}_{zn}}{\tilde{F}_{z1}} = \frac{1}{n^3} \frac{\left(1 - e^{-\frac{2n\pi}{l} \Delta}\right) \left(1 - e^{-\frac{2n\pi}{l} \Gamma}\right) e^{-\frac{2n\pi}{l} x_0}}{\left(1 - e^{-\frac{2\pi}{l} \Delta}\right) \left(1 - e^{-\frac{2\pi}{l} \Gamma}\right) e^{-\frac{2\pi}{l} x_0}}$$

Where $F_{z1} \neq 0$. The ratio above is about the order of $1/n^3$. So, we can estimate the effect of the first order non-zero higher harmonics in the motor. In addition, according to the allowable amplitude of the force ripples, wavenumber n can be determined. For example, the rms force ripples due to variation of the permanent magnet properties, manufacturing tolerances and power amplifier offsets is equal to 0.06% [11]. If the force ripples caused by higher harmonics are needed to be less than that, n should be larger than 12. And if the force ripples caused by higher harmonics are needed to be ten times less than 0.06%, n should be larger than 22.

According to the analysis results above, increasing the order of the first order non-zero higher harmonic can reduce the force ripples effectively. Further, the influence of geometry parameters s and t on the order of the first order non-zero higher harmonic is researched.

$$n = ps + 1 = qt + 1 \quad (4)$$

$$n = ps - 1 = qt + 1 \quad (5)$$

If the order of certain higher harmonics of force can meet neither of the two equations above, the amplitude of this higher harmonics is zero. Therefore, the solutions of the two equations is important. We know, there're infinitely many solutions of the two equations, and the least positive integer of them is the order of the first

order non-zero higher harmonics. Let d be the maximum common divisor of s and t . The least positive integer solution of equation (4) is

$$n_{a1} = \frac{st}{d} + 1$$

When d larger than 2, there's no non-negative integer solution of equation (5). So, n_{a1} is the order we need. Otherwise, the least positive integer solution of equation (5) is less than n_{a1} . And it is what we need. The first order non-zero higher harmonics of force varied with parameters s and t is listed in table 1. Reasonable parameters s and t can be determined according to n needed.

TABLE 1. The first order non-zero higher harmonics of force varied with parameters s and t .

$t \backslash s$	2	3	4	5	6	7	8	9	10	11	12
2	3	5	3	9	5	13	7	17	9	21	11
3	7	4	7	4	7	13	7	10	19	10	13
4	5	5	5	9	5	13	9	17	9	21	13
5	11	11	11	6	11	6	31	26	11	21	11
6	7	7	7	19	7	13	7	19	19	43	13
7	15	8	15	29	29	8	15	8	29	43	71
8	9	17	9	9	17	41	9	17	9	65	25

The clearance between neighboring windings is researched to eliminate certain order non-zero higher harmonics. Expression (2) is a sine function varied with n . There's an appropriate value of the clearance to meet the two expressions below.

$$\sin\left(\frac{n}{s}\pi - \alpha n\pi\right) = 0$$

$$\sin\left(\frac{1}{s}\pi - \alpha\pi\right) \neq 0$$

That is

$$\alpha = \frac{1}{s} - \frac{m}{n}$$

where m is a non-negative integer. By this method, the amplitude of the n th higher harmonics is zero and the amplitude of the fundamental is not zero. The aim of eliminating certain order non-zero higher harmonics comes true. Further, the larger α is, the smaller the amplitude of the fundamental is. So, we need the largest m . For example, a motor with $t = 4$ and $s = 12$ is researched. The first order non-zero higher harmonics of force in this motor is 13. Let m to be 1, and α will be 0.006. In this case, the amplitude of the 13th higher harmonics is zero, and the first order non-zero higher harmonics becomes 25. Another example, a motor with $t = 4$ and $s = 6$ is researched. The

first order non-zero higher harmonics of force in this motor is 5. There's no non-negative integer solution of m . So, if this method will be applied to eliminate the first non-zero higher harmonics of force in the type of ILPMLSM researched in this paper, the relationship between s and t should meet $s \neq qt + 2$, where q is a non-negative integer.

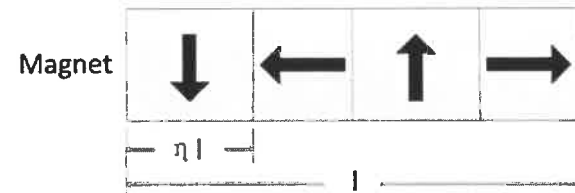


FIGURE 3. The structure of the magnet array that the length of the z-directed magnetization magnet is different from the length of the x-directed magnetization magnet when $t = 4$.

In the same way, appropriate clearance between neighboring magnets can eliminate certain order non-zero higher harmonics of force. But, applying it to eliminate higher harmonics, assembly will be difficult to ensure the magnitude of the clearance. There's another method that can both solve the two problems. We focus on the difference between the length

of the z-directed magnetization magnet and the x-directed magnetization magnet when $t = 4$. The length of the x-directed magnetization magnet is assumed to be η times of pitch. The structure of this magnet array is shown in figure 3. The expression (3) is converted into

$$g(\eta) = \frac{2\sqrt{2}}{\pi} \sin\left(\frac{\pi}{4} + \eta n \pi\right)$$

Similar to the clearance between neighboring windings, there's an appropriate value of the length of the x-directed magnetization magnet to meet the two expressions below.

$$\sin\left(\frac{\pi}{4} + \eta n \pi\right) = 0$$

$$\sin\left(\frac{\pi}{4} + \eta n \pi\right) \neq 0$$

That is

$$\eta = \frac{m - \frac{1}{4}}{n}$$

where m is a positive integer. Further, when parameter η meets the expression below, the amplitude of fundamental is largest.

$$\eta = \frac{n-2}{4n} \text{ or } \frac{n+2}{4n}$$

VERIFICATION

The results are validated by comparison of the ILPMLSM with FEA. The forces of three ILPMLSMs are calculated by analytical solution and FEA. The first of the three motors is original motor. The second of the three is the motor which magnet array has been optimized to eliminate higher harmonics. And the third of the three is the motor which windings and magnet array have been optimized to eliminate higher harmonics. The parameters of the motors are the following: number of phases, $s = 6$; number of magnets in one period, $t = 4$; pitch, $l = 72\text{mm}$; depth, $w = 50\text{mm}$; length, $L = 2l$; winding thickness, $\Gamma = 8\text{mm}$; magnet array thickness, $\Delta = 5\text{mm}$; air gap thickness, $x_0 = 1\text{mm}$; remanence, $\mu M_0 = 1.45\text{T}$. There're 356 turns of coil in each winding of the first and second motors, and 329 turns of coil in each winding of the third one.

The first three orders non-zero higher harmonics of the force are 5, 13 and 17 in the original motor. Optimization of the length of the x-directed magnetization magnet eliminates the 5th order higher harmonics, and optimization of the clearance between neighboring windings eliminates the 13th order higher harmonics. According to the analysis in the section above, the optimized parameters are the following: $\varepsilon =$

0.05; $\alpha = 0.013$. The forces and force ripples of three ILPMLSMs calculated by analytical solution and FEA are shown in figure 4 and table 1. The methods proposed in the section above are effective of eliminating certain higher harmonics to reduce force ripples in the type of ILPMLSM researched in this paper.

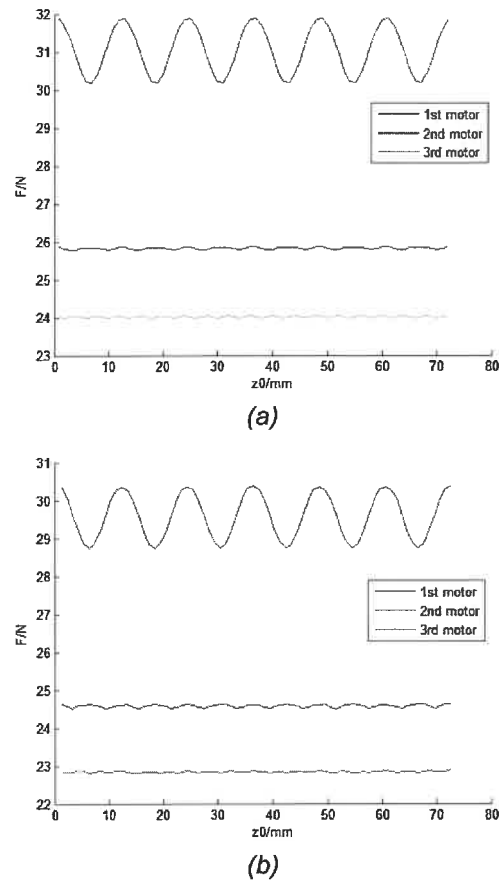


FIGURE 4. Forces of three ILPMLSMs calculated by analytical solution and FEA. (a) Force of three motors calculated by analytical solution. (b) Force of the three motors calculated by FEA.

CONCLUSIONS

The main purpose of this paper is to provide a general design and analysis method for a type of ILPMLSM with little force ripples. A novel design theory of ILPMLSM is proposed, which easily meets the requirement of little force ripples by eliminating higher harmonics as more as possible. The expression of higher harmonics of force is derived via magnetic vector potential and Maxwell stress tensor. The analysis result shows that the first order non-zero higher harmonic is the main component of the force

ripples caused by higher harmonics. Further, methods to eliminate certain order non-zero higher harmonics of force are proposed: appropriate clearance between neighboring windings, appropriate clearance between neighboring magnets and appropriate length of the x-directed magnetization. The results are effective validated by comparison of the ILPMLSM with FEA.

TABLE 2. The forces and force ripples of three ILPMLSMs calculated by analytical solution and FEA.

	force/N		force ripples/%	
	analytical solution	FEA	analytical solution	FEA
1st	31.05	29.56	2.65	2.73
2nd	25.81	24.57	0.16	0.25
3rd	23.95	22.83	0.10	0.14

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A FLEXURE-BASED ROLL-TO-ROLL MACHINE FOR FABRICATING FLEXIBLE PHOTONIC DEVICES

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In this paper, we present the characterization and precision control of a flexure-based roll-to-roll (R2R) printing system as well as the adaptation of a Microcontact Printing (MCP) process to the R2R platform for fabricating submicron resolution flexible photonic devices, including 600 nm optical gratings and various gold and silver electrodes for organic electronic devices. High-resolution continuous R2R printing is enabled by the fully automated flexure-based R2R machine with multi-axis misalignment correction capability.

ULTRA PRECISE R2R PRINTING

Compatibility with R2R processes has been the main driving force for flexible electronics and photonic devices; however there is a common misperception that this high throughput, low-cost process produces devices of lower resolution and quality. This is not necessarily true if contact printing techniques such as Microcontact Printing [1] or Nanoimprint [2-3] are implemented to properly designed R2R platforms. In fact, the printing resolution can be better than lithographic processes as contact printing is not limited by the diffraction of light. To date, this has not been realized as current state-of-the-art R2R systems, e.g. gravure printing and flexographic printing, only have ~15 micron print resolution. This is because conventional bearings are used in these systems to guide the printing motion and lack the required repeatability and precision that lead to non-uniform pressure distribution and pattern distortions.

To utilize a R2R process for precision printing, one must address the machine's repeatability and accuracy issue. A R2R machine must possess (1) nanometer repeatability, (2) submicron level positioning precision, and (3) better than 0.1 N force control in order to create patterns with submicron resolution.

MECHANICAL CHARACTERIZATION

As shown in Figure 1, we have developed a flexure-based R2R printing system for submicron resolution printing [4]. Nanometer repeatability and multi-axis error correction capabilities are achieved through the flexure-based positioning system. PID control is implemented in all sub-modules of the R2R system, including the positioning stage, the web tension controller, and the web guide system [4].

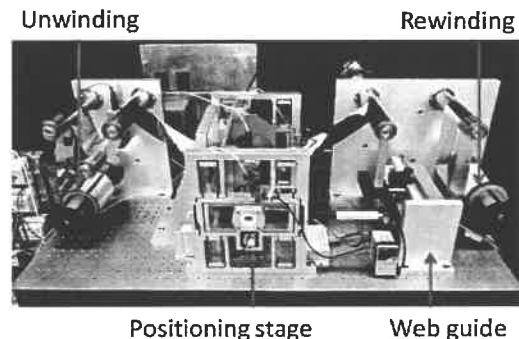


FIGURE 1: Prototype of the flexure-based roll-to-roll printing system

PARASITIC MOTION CHARACTERIZATION

Figure 2 shows the CAD model of the core positioning stage, where two monolithic compliant stages are installed to support the two ends of a print roller with air bearing and integrated load cells.

Although each monolithic X-Y stage presents a completely decoupled design, unwanted motion coupling at micron to submicron level is still present due to limited manufacturing tolerance and imperfect constraint and loading conditions. In addition, motion coupling across different X-Y stages is present due to a rigid connection of print roller.

To further improve positioning precision, we devised four experiments to understand and characterize the motion coupling behavior between two X-Y compliant stages. Since

parasitic motions in a flexure mechanism are highly repeatable (~20nm), these results can be used to eliminate parasitic motions with a properly designed controller.

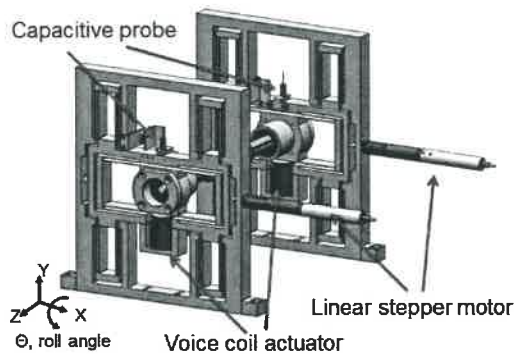


FIGURE 2. CAD model of the positioning stage with integrated actuators and sensors

As shown in Figure 2, both X-Y stages are driven by linear stepper motors (M230.10S, PI) in the X direction, and voice coil actuators in the Y direction (NCC03-15-050-2X, H2W). The position sensing is performed by capacitance probes (C30/CPL290, Lion Precision). To reduce the external disturbances such as ground vibration, all experiments were performed on an air table (Smarttable Top/S-2000 isolators, Newport) so as to reduce the ground vibration to within 40 nm.

In the first experiment, stage 1 was commanded to move in the Y direction (Y_1) for $\pm 200 \mu\text{m}$, and the coupling Y motion on stage 2 (Y_2) was measured. In the second experiment, stage 1 was commanded to move in the X direction (X_1) for $\pm 90 \mu\text{m}$, and the coupling X motion on stage 2 (X_2) was measured. The results are shown Figure 3A and 3B respectively. The results suggest a strong, yet linear, motion coupling effect, which may be largely removed by a correction matrix with open-loop control or completely removed by a closed-loop controller.

The relationship between Y_1 and Y_2 as well as X_1 and X_2 can be described by equation (1) and (2) respectively:

$$Y_2 = 0.02 + 0.336 * Y_1 \quad (1)$$

$$X_2 = -0.02 + 0.402 * X_1 \quad (2)$$

In the third experiment, stage 1 was commanded to move in the Y direction (Y_1) for

$\pm 200 \mu\text{m}$, and the coupling X motions on both stage 1 (X_1) and stage 2 (X_2) were measured. In the fourth experiment, stage 1 was commanded to move in the X direction (X_1) for $\pm 90 \mu\text{m}$, and the coupling Y motions on both stage 1 (Y_1) and stage 2 (Y_2) were measured. The results are presented in Figure 4A and 4B respectively. These results prove that each monolithic compliant stage has excellent motion decoupling capability and that motions from different direction cause parasitic motions on the scale of 100-200 nm. More importantly, these parasitic motions are relatively linear and can be eliminated by proper control strategy.

Linear relationships between Y_1 and X_1/X_2 as well as X_1 and Y_1/Y_2 can be described by equation (3)-(6) respectively:

$$X_1 = 4.12e^{-3} + 3.29e^{-3} * Y_1 \quad (3)$$

$$X_2 = 4.15e^{-4} + 9.52e^{-4} * Y_1 \quad (4)$$

$$Y_1 = -5.32e^{-4} - 4.06e^{-4} * X_1 \quad (5)$$

$$Y_2 = 8.7e^{-5} + 1.53e^{-3} * X_1 \quad (6)$$

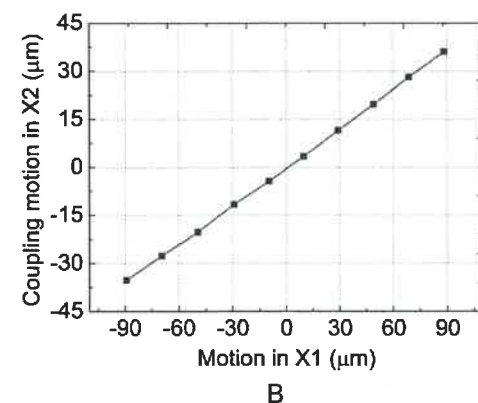
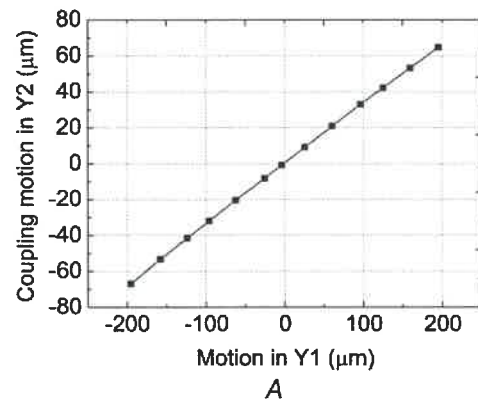


FIGURE 3. Cross-stage motion coupling behavior between the two X-Y stages in the Y direction (A) and X direction (B)

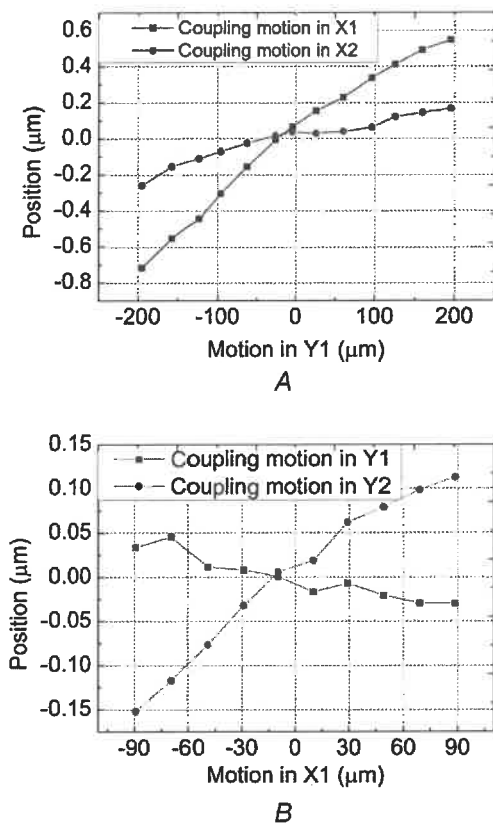


FIGURE 4. Cross-axis motion coupling behavior in Y direction (A) and X direction (B)

CONTROL STRATEGY

A PID controller that precisely controls the contact force and yaw/roll angle of the print roller was developed based on the results of the motion coupling experiments. The goal of this controller is to set a uniform contact force at any desired value (0-15N) with minimized settling time and no overshoot. To reach high force uniformity, the yaw angle (θ_y) and roll angle (θ_x) of the print roller should be minimized.

An iterative approach was adopted as our control strategy to minimize the yaw and roll angle. First, a target positioning (i.e. printing force) signal is sent to two Y actuators in both stages, where each stage is controlled by an independent PID controller. If the measured X/Y positions are different, new target positioning signals of X and Y axis for each stage will be calculated based on the linear relationships presented in Equation (1) – (6) and set to be the new target signals for individual controllers. For example, if roll angle errors are present, the new target signals will command the Y actuators to generate a torque on the print roller to reduce

the angular error. After 3-5 iterations, both the contact force and the yaw/roll angle will settle on the target value, thereby achieving motion decoupling.

R2R PRINTING EXPERIMENTS

We devised two experiments to verify the roll angle and contact force control capability. In the first experiment, the print roller was commanded to move down and up in 2 N force steps. The results are shown in Figure 5, where throughout the experiment contact force can be controlled within ± 0.02 N without any overshoot.

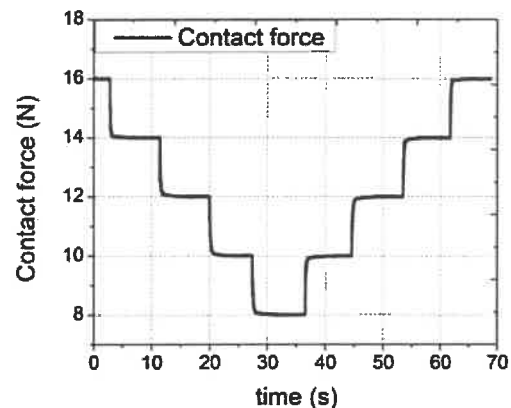


FIGURE 5. The close-loop contact force experiments for a 2 N force steps

In the second experiment, we recorded the average contact force and position of the print roller for one revolution (360°) with and without the PID controller. Figure 6 shows the printing force data with a target printing force of 15N. The closed-loop results show consistent precision throughout the experiments (± 0.05 N), while the open-loop results show significant deviation due to roller eccentricity that cannot be avoided at submicron level. More experiments indicate slowing down the printing speed can further improve the force and angular control accuracy.

STAMP PREPARATION

To adapt the MCP process for R2R operation, we replaced the normal substrate with a 4" wide metal coated PET roll. For stamp preparation, PDMS stamps are first fabricated by standard MCP procedures [1], following which the stamp is bonded to a glass cylinder that is then securely mounted to an air-expandable, motor-driven print roller shaft. Figure 7 shows an image of the glass cylinder with a bonded PDMS stamp. We have also developed new etching

recipes for both gold and silver to be compatible with the fast R2R process (etch time < 1 ms).

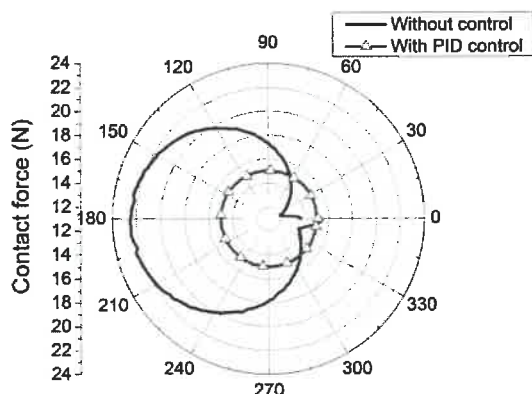


FIGURE 6: Open-loop and closed-loop printing force data

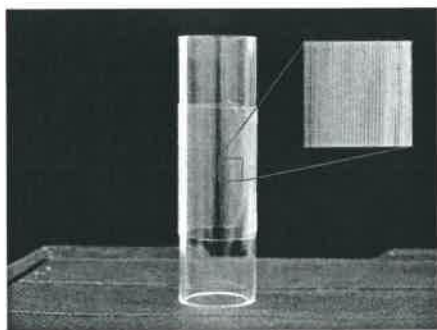


FIGURE 7: PDMS stamp securely bonded to a glass cylinder by oxygen plasma treatment

DEVICE FABRICATION

Figure 8 and 9 show various metal patterns printed by the R2R system on a 4" PET web. Figure 8A shows an image of flexible gold electrodes around 2" x 2" in size (cut from a 4" web). Figure 8B shows four SEM images of both gold and silver square/hexagonal grids with line widths ranging from 20-50 microns. Figure 9 shows the SEM image of a R2R patterned optical grating with a line width of 600 nanometers, printed on the 4" web. These exciting results present the highest resolution patterns produced by a large area R2R system for the first time

CONCLUSION

We have developed a flexure-guided R2R machine and achieved nanometer-level positioning resolution and uniform force control over a 4" web within $\pm 0.05\text{N}$. By adapting MCP, we have, for the first time, scaled up the MCP process over a large area (4" PET web) and achieved 100 nanometer print resolution. The printed metal electrodes were used to fabricate

various photonic devices including an organic photovoltaic cell and photodetector.

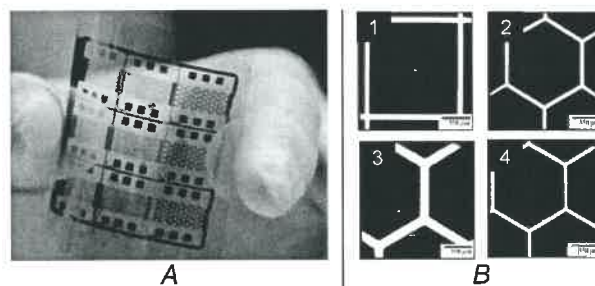


FIGURE 8: R2R printed samples on PET web (A); SEM images of printed gold (1 & 2) and silver (3&4) electrodes; scale bars in 1-4 are 350 microns



FIGURE 9: R2R printed sample of optical grating (gold lines); the line width in the SEM image is 600 nanometers; scale bar = 6 micron

ACKNOWLEDGEMENT

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A NOVEL QUASI-ZERO MAGNETIC BEARING IN LONG STROKE OF RETICLE STAGE

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INSTRUCTIONS

In many industrial equipment, workpiece or workpiece-stage needs to be driven and positioned in 6 degree of freedom, such as the wafer stage and the reticle stage in lithography. Conventional wafer stage and reticle stage, which are supported by machine guide and air bearing, are complicated and require high environmental control. Magnetic bearing with gravity compensators can achieve their levitation with simple structures, excellent environmental adaptability and low power consumption.

This document provides a novel quasi-zero magnetic bearing in long stroke of reticle stage in step & scan lithography. This magnetic bearing consists of the actuators for adjusting the position and attitude of five degrees of freedom except the scanning direction and the magnetic gravity compensators. The actuators and the magnetic gravity compensators are optimized and verified by finite element simulation and experiment. The magnetic gravity compensators achieve a quasi-zero stiffness in the range of $\pm 1\text{mm}$ of stepping and vertical directions.

THE STRUCTURE OF MAGNETIC BEARING

The magnetic bearings are symmetrically disposed below the long stroke's corners. Its structure is presented in Figure 1. The actuators are moving-coil linear motors, which provide the forces in stepping and vertical directions, for adjusting the position and attitude of five degrees of freedom except the scanning direction. Magnetic gravity compensators are special magnet array. The repulsion between the permanent magnet array of moving-coil linear motor and the magnetic gravity compensator is used to compensate the gravity of moving stage.

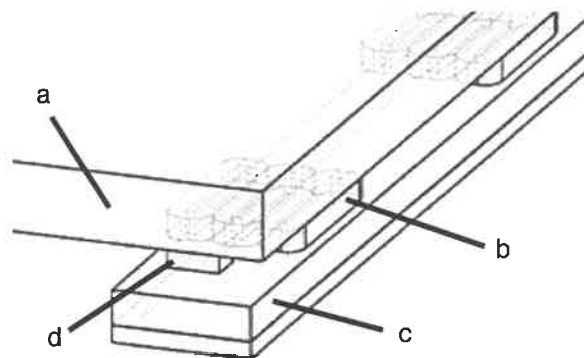


FIGURE 1. The structure of magnetic bearing
a. long stroke; b. coils of actuator; c. permanent magnet array of actuator; d. magnetic gravity compensator.

ACTUATOR

Structure of the Actuator

Figure 2 shows the structure of actuator. The actuator is a three-phase moving-coil linear motor¹. Its stator is a one-dimensional Halbach array composed of five magnets. In order to achieve a large stroke in the scanning direction, the length of the permanent magnet array in the scanning direction is one meter.

In order to obtain sufficient thrust, the rated current of the actuator is 10A. When it works, a large amount of heat is generated. The heat must be taken away to avoid changing the work environment of lithography. So, a cooling structure is designed for this purpose.

Optimization of the Actuator

Due to space constraints, the dimensions of the actuator in stepping and vertical directions are determined value. In order to minimize the power consumption of actuator, the dimensions of the magnet with different magnetization directions, and the thickness of coil are optimized. After

optimization, when the current is 1A, both the maximum step and vertical thrust in the equilibrium position of the actuator achieve 10.7N in the finite element simulation of Two-dimensional model.

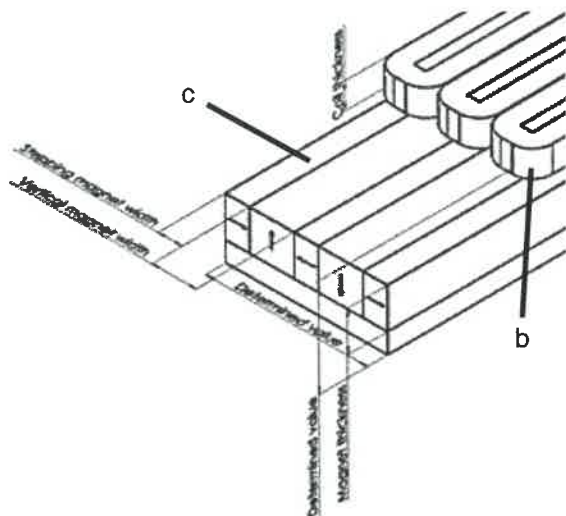


FIGURE 2. The structure of the actuator
b. coils of actuator; c. permanent magnet array of actuator.

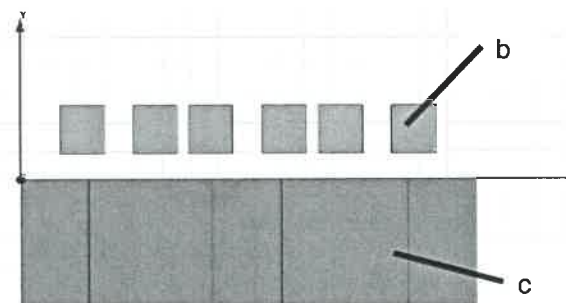


FIGURE 3. The two-dimensional model of the actuator
b. coils of actuator; c. permanent magnet array of actuator.

Verification of the Actuator

When we complete the design of the actuator, several actuators and some experimental devices were manufactured for verifying the thrust and cooling performance of actuators.

Based on past experience of motor simulation, the experimental results is about 80 percent of the finite element simulation results of Two-dimensional model. In the test, both the maximum step and vertical thrust in the equilibrium position of the actuator achieve 8.5N when the current is 1A, 79 percent of the finite element simulation results, in line with

expectations. Figure 4 shows the experimental device.

In reticle stage, the actuators of magnetic bearing are not always working on rated current. We estimate that the average current is 3A. The surface temperatures of actuators working on average current are tested by thermal imager. The maximum temperature rise is less than 2.4°C. Figure 5 and 6 show the measurement results.

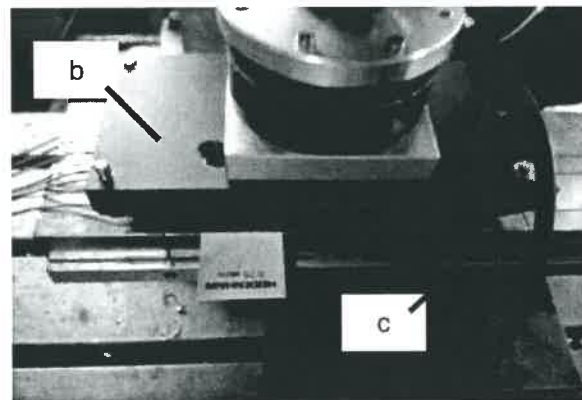


FIGURE 4. The experimental device of the actuator
b. coils of actuator; c. permanent magnet array of actuator.

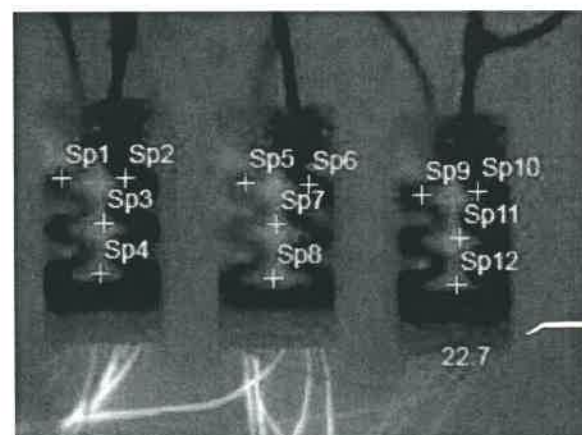


FIGURE 5. Measurement results by thermal imager

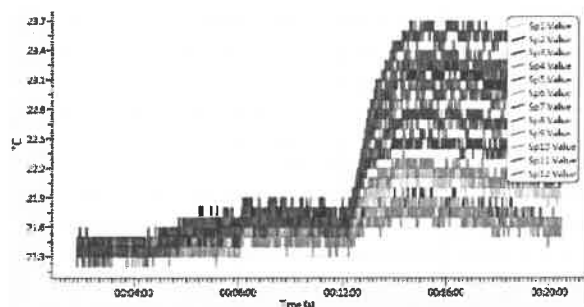


FIGURE 6. The surface temperatures of actuators working on average current

MAGNETIC GRAVITY COMPENSATOR Structure of the Magnetic Gravity Compensator

Figure 7 shows the structure of the magnetic gravity compensator. It is mounted above the permanent magnet array of actuator, and consists of three magnets with different magnetization directions. The direction of the force between the middle magnet and the permanent magnet array is different from the ones between the both sides of the magnet and the permanent magnet array. The resultant force between the magnetic gravity compensator and the permanent magnet array is used to compensate the gravity of moving stage.

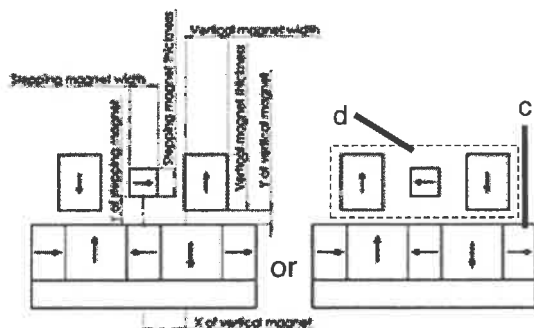


FIGURE 7. The structure of the magnetic gravity compensator (After optimization, the left structure is better.)

c. permanent magnet array of actuator; d. magnetic gravity compensator.

Optimization of the Magnetic Gravity Compensator

To obtain a stable compensation force when long stroke had a small stroke $\pm 1\text{mm}$ movement in the stepping and vertical directions, the stiffness of magnetic gravity compensator in the stroke $\pm 1\text{mm}$ is expected to be as small as possible. Zero stiffness is ideal.

In order to calculate the stiffness, the numerical integral analytical solution of the force between the magnetic gravity compensator and the permanent magnet array is established by equivalent magnetic charge method², equivalent current method³ and Ampere equation. Because the numerical integration solution cannot be optimized, Multi-island genetic algorithm is used to find the optimal solution.

After optimization, the magnetic gravity compensator achieves a quasi-zero stiffness near the equilibrium position. The stiffness by the analytical solution is less than 0.5N/mm , and the stiffness by the finite element simulation solution is less than 3N/mm . Figure 8 shows the compensation force curve by the analytical solution and the finite element simulation solution.

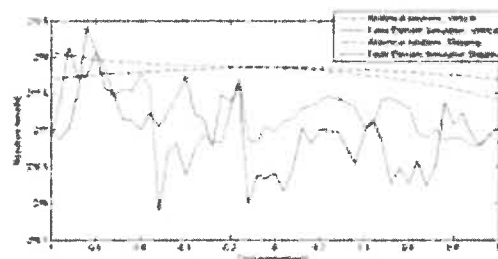


FIGURE 8. The compensation force curve by the analytical solution and the finite element simulation solution

Verification of the Magnetic Gravity Compensator

Three magnets are bonded to the shell, they together constitute the magnetic gravity compensator. A force sensor in 6 degree of freedom mounted on the magnetic gravity compensator is used to test the forces and torques.

The forces and torques in 5 degree of freedom except vertical are almost zero. The vertical force is about 251N , and the stiffness is less than 4.8N/mm , in line with expectations. Figure 9 shows the experimental device for verifying the quasi-zero stiffness of the magnetic gravity compensator. Figure 10 shows the experimental results.

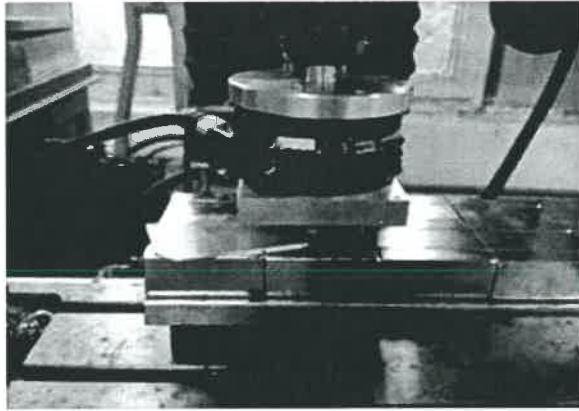


FIGURE 9. The experimental device for verification the quasi-zero stiffness of the magnetic gravity compensator

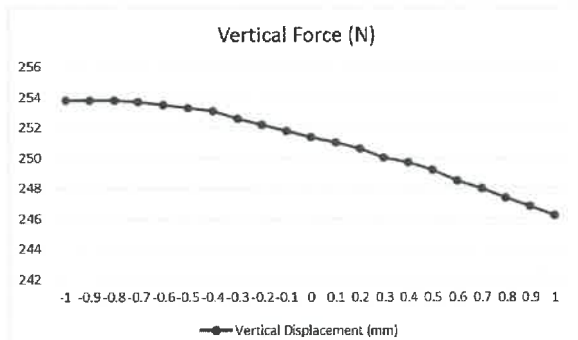


FIGURE 10. The experimental results of the magnetic gravity compensator

THE VERIFICATION OF MAGNETIC BEARING

A reticle stage, which used the magnetic bearing provided herein, has been designed and manufactured. Its commissioning is in progress. The reticle stage is expected to achieve maximum acceleration of 70m/s^2 , MA of 2nm and MSD of 5nm.

SUMMARY

This quasi-zero magnetic bearing can provide a stable compensation force when long stroke had a small stroke $\pm 1\text{mm}$ movement in the stepping and vertical directions, and adjust the position and attitude of five degrees of freedom except the scanning direction. It can help reticle stage realize high speed and high acceleration precision motion.

ACKNOWLEDGEMENTS

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SPECTRAL IMAGING METROLOGY USING CHROMATIC ABERRATION

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1. INTRODUCTION

Spectral imaging has found application in many fields such as remote sensing [1], biomedical imaging [2], agriculture [3], etc. Spectral imaging is measuring the intensity of light as a function of wavelength, λ , for any point (x,y) on the object. The results is a three-dimensional data cube, which gives intensity as a function of (x,y,λ) . In conventional spectral imaging techniques, there is a tradeoff between spectral and spatial resolutions. In this paper, a spectral imaging technique is proposed based on the images obtained from a dispersive imaging system. In this spectral imaging method, the spectral and spatial resolutions are decoupled and can be improved both at the same time.

Shape from focus (SFF) method [4] is applied to images obtained from a dispersive lens to compute the data cube containing spectral images. SFF is a popular method used for depth recovery problem. In this method several images are taken from a three-dimensional object at different distances from the imaging lens and then an algorithm is used to obtain the sharpness of images versus distance from imaging lens for each pixel. Depth information is determined based on the location of focus or maximum sharpness. Similar scenario happens if spectral information of a two-dimensional object is of interest and imaging is done by a dispersive imaging system. A dispersive imaging lens system has a focal volume instead of focal plane and will spread the wavelength information of the object over the optical axis, the z-axis of image volume. Images taken at different locations of this axis contain unique wavelength information of the object at a given wavelength. SFF is applied to images of such system in order to calculate the sharpness of the images versus wavelength, instead of distance, for each pixel. This way the wavelength

information of the whole object is determined and the spectral images are formed.

Our spectral imaging system will be a powerful tool for optical metrology of contaminated surfaces or surfaces with complex composition. Since different materials have different wavelength signatures, images of contamination and the original material can be distinguished by use of spectral imaging. It also has the capability to be used for high speed surface metrology. A conventional chromatic confocal microscope [5] requires a point by point scanning of the object. In comparison, our spectral imaging system is much faster because it gives the spectral images of the whole object at once and doesn't need point by point scanning.

This paper is written as follows: in next section, a new spectral imaging system is described and in section 3, a dispersive lens system, which is a critical part of our spectral imaging system, is presented. Applying SFF to obtain images and forming the data cube is explained in section 4. Results of proposed system are given in section 5. Conclusive remarks are given in section 6.

2. DISPERSIVE SPECTRAL IMAGING SYSTEM

Our novel spectral imaging system includes a highly dispersive imaging lens system (see figure (1)). The idea is to make use of the chromatic aberration of such imaging system to separate and disperse spectral information of object over the focal volume of the lens system. By taking several images of the object at different planes of this volume, wavelength information of the object can be determined. For instance, if the image plane is located at the focal plane of the red wavelength, all the information related to red wavelength would be in-focus on this image plane and information related to other wavelengths would be out-of-focus. Based on this fact and by mean of SFF

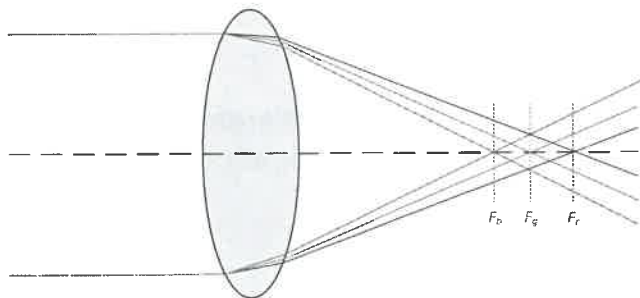


FIGURE (1) *Instead of a focal plane, dispersive imaging system has a focal volume which is wavelength dependent.*

method, spectral images can be extracted from proposed spectral imaging system. What we need is to design an imaging lens system, which is dispersive, i.e. has large chromatic aberration. Then several images of the object produced by this lens system are taken and synthesized by SFF. For each pixel of the image, SFF gives us the locations of focus and each location of focus corresponds to a specific wavelength.

As mentioned before, the spectral and spatial resolutions are decoupled in this spectral imaging system. Spectral resolution depends on the number of images taken over the focal volume of the lens system. However, the spatial resolution depends on the resolution of the camera. Since spectral and spatial resolutions are decoupled, both can be improved simultaneously.

3. DISPERSIVE IMAGING LENS SYSTEM DESIGN

As can be seen from figure (1), dispersive imaging lens system is a key component in our spectral imaging technique. Designed imaging lens system should have large chromatic aberration while other types of aberration are minimized. The larger is the chromatic aberration the better wavelength resolution can be achieved. At the same time, other types of aberration are desired to be eliminated to maintain the quality of images. In order to satisfy mentioned requirements, a triplet lens system was designed. Triplet design gives us enough degrees of freedom to achieve our objectives.

The design method introduced in [6] was followed here with one difference. While designing an imaging system, designers normally try to eliminate chromatic aberration of the system. A constraint can be imposed on the design to make sure the focal planes of blue and red wavelengths coincide. However, in our case

we are making advantage of the chromatic aberration of the imaging system. Not only we don't want the red and blue focal points to coincide but we want them to be separated as much as possible to improve the spectral resolution of the spectral imaging system. Same constraints as [6] were used here except for the constraint that makes the distance between blue and red focal points equal to zero. Instead of zero, this constraint was imposed to be equal a non-zero constant, which determines the wavelength resolution of spectral imager. The larger is this number, the farther away are the blue and red focal planes and the better spectral resolution can be achieved.

4. CALCULATING SPECTRAL IMAGES BY APPLYING SFF

SFF is a well-known method used for depth recovery problem [4]. The basic idea behind the method is that if we know the location of focus, we can figure out what was the object distance using the lens formula. Several images of a three dimensional object are taken at different distances from a lens system and the location of focus is determined for each pixel. Various criteria functions can be defined in order to measure the sharpness of an image [7]. The value of the criteria function is calculated for all taken images at each pixel and the maximum of this function determines the location of focus.

We are dealing with a similar problem to depth recovery in our case of spectral imaging. The difference is that the object is two-dimensional and it is the wavelength information of each pixel that determines the location of focus not the depth.

In our spectral imaging system, we have a two dimensional object and an imaging system, which its imaging properties are highly wavelength dependent i.e. is highly dispersive. If a pixel has information at a specific wavelength, that pixel would be in-focus on image plane of the lens system corresponding to that specific wavelength. For example, assume that we are looking for all the pixels containing information at specific wavelength of λ_0 and the image plane corresponding to this specific wavelength is at distance d_0 from the lens system. Any pixel which its corresponding focus measure function has a maximum at d_0 , contains information at λ_0 . A pixel can have information at several wavelengths. In that case focus measure function has multiple maximums.

Once the imaging lens system with large chromatic aberration is designed, several

images are taken from the object at different locations between red and blue image planes. A criteria function is chosen and its values are calculated over all image planes. To extract the specific wavelength information, we look at the image plane corresponding to that specific wavelength and look for pixels which are in focus on that plane.

5. RESULTS

A dispersive imaging lens system is designed in Zemax based on the method of section 3. The layout of final design is shown in figure (2). The triplet lens system shown in figure (2) has an effective focal length of 50 mm and image space f-number of 2.5. The distance between F (486.13 nm) and C (656.27 nm) image planes of designed system at paraxial magnification of -1 is 4.52 mm. Considering diffractive depth of focus [6] as limiting factor for the maximum number of taken images, the best spectral resolution that can be achieved by this system is approximately 4.3 nm. In order to show the functionality of spectral imaging system, object in figure (3) was chosen. The object contains information at three different wavelengths. Dispersive properties of the lens system bring each wavelength in-focus at different location. Images were taken at 50 μ m steps between F and C image planes and were analyzed by SFF technique. The criteria function used to measure the sharpness at each image was grey-level variance and 9-by-9 window was used around each pixel [7]. The result is shown in figure (4). Pixels containing information at blue color became in focus at a distance of 85.8 mm from the last surface of the lens system. This distance is 88.9 mm and 90.3 mm for green and red color, respectively. Since the dispersive properties of the lens system is thoroughly known, we can say each distance corresponds to what wavelength. The calculated spectral

images of the final data cube are shown in figure (5).

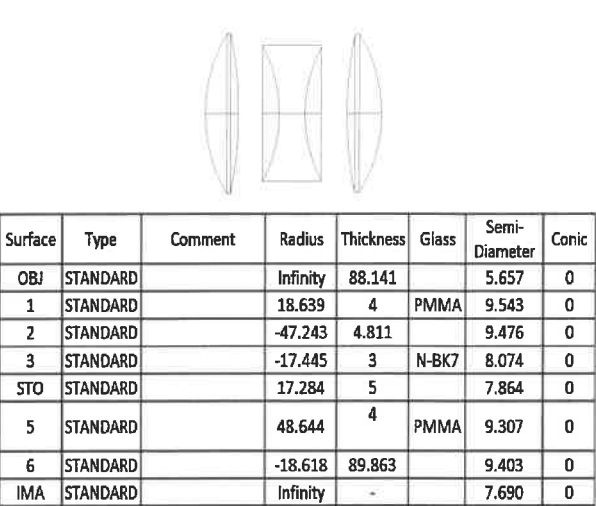


FIGURE (2) Layout of the imaging lens system along with summary of surface data

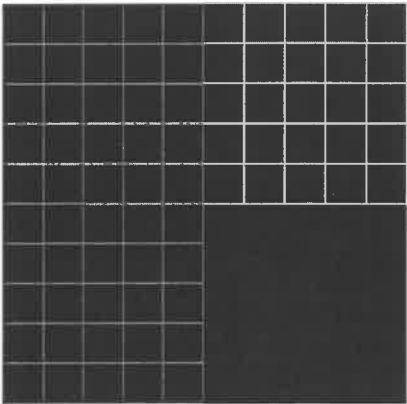


FIGURE (3) Object contains information at three different wavelengths

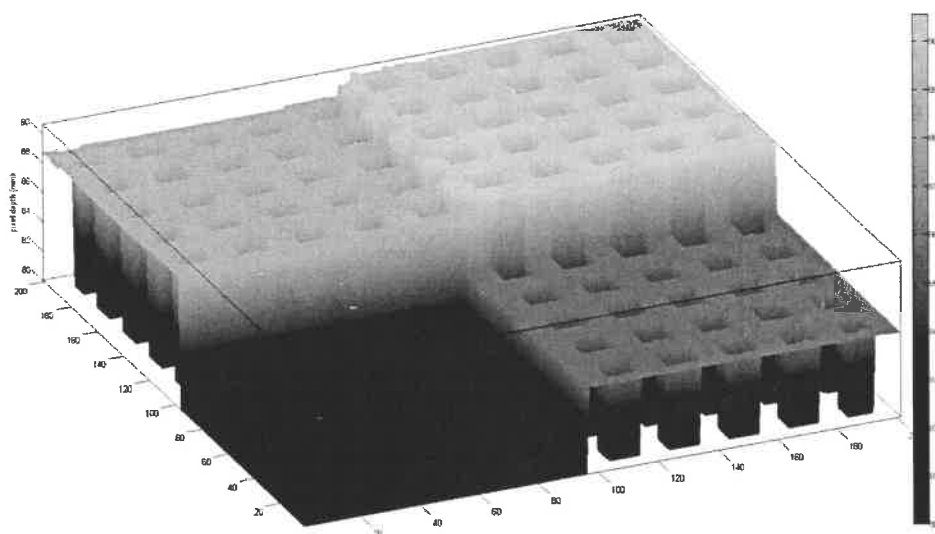


FIGURE (4): SFF is applied to images of object shown in figure (2) obtained from designed system shown in figure (3). Three different heights correspond to three wavelengths in object. Pixels containing information at blue, green and red color became in focus at of 85.8 mm, 88.9 mm and 90.3 mm distance from the last surface of the lens system, respectively.

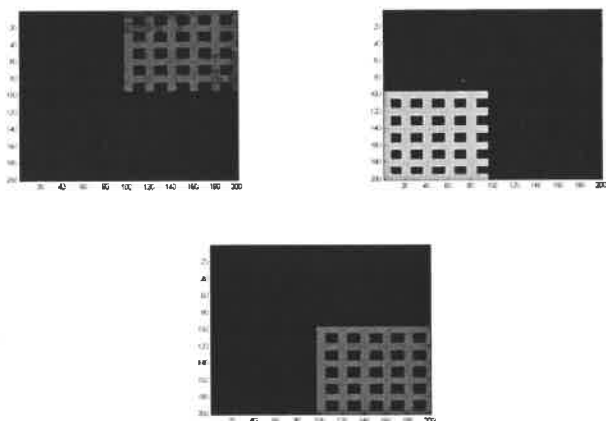


FIGURE (5): Spectral images of the object in figure (3). These images form a data cube which is the output of spectral imaging.

Results from an experimental setup are also presented here. A triplet lens system was used along with a CCD camera to capture images of object shown in figure (6). The object consists of three slits and was illuminated by a red LED positioned behind the right slit and by a blue LED positioned behind the left slit. 106 images were taken at different distances from the lens system in 0.005 inch steps. Size of each image



FIGURE (6): Object used in experimental setup. A red LED was positioned behind the right slit on the object and a blue LED was positioned behind the left slit

was 1280×1024 . Same as before, images were analyzed by SFF method and the spectral images are shown in figure (7). Since the red (blue) LED was positioned behind the right (left) slit, the spectral image corresponding to red (blue) wavelength fades by moving toward left (right) direction. Most part of the middle slit is illuminated by both LEDs and thus criteria